ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

THE EFFECTS OF CYLINDER DEACTIVATION ON ENGINE LUBE OIL CONSUMPTION

M.Sc. THESIS

Celal Gürsen ALTIN

Department of Mechanical Engineering

Automotive Programme

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İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

SİLİNDİR DEAKTİVASYONUNUN İÇTEN YANMALI MOTORLARDA YAĞ TÜKETİMİNE ETKİLERİ

YÜKSEK LİSANS TEZİ

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To my beloved spouse,

FOREWORD

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June 2013 Celal Gürsen ALTIN
Mechanical Engineer

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ABBREVIATIONS

ASY	: Assembly
ATS	: Anti-Thrust Side
AVL	: Anstalt Für Verbrennungskraftmaschinen List (Company)
BDC	: Bottom Dead Center
CCV	: Closed Crankcase Ventilation
CFD	: Computational Fluid Dynamics
CI	: Compression Ignition
CVCP	: Continuously Variable Cam Phaser
DPF	: Diesel Particulate Filter
ECU	: Engine Control Unit
EGR	: Exhaust Gas Recirculation
EPA	: U.S. Environmental Protection Agency
EVAP	: Evaporation
FEV	: Engineering Problem Solution And Consultancy Company
FIP	: Fuel Injection Pump
F-TDC	: Firing Top Dead Center
GM	: General Motors
GMEP	: Gross Mean Effective Pressure
HLA	: Hydraulic Lash Adjuster
HTC	: Heat Transfer Coefficient
IMEP	: Indicated Mean Effective Pressure
Jap.10-15	: Japanese 10-15
LNT	: Lean NOx Trap
LOFT	: Left Oil Film Thickness
LOC	: Lube Oil Consumption
M.Sc.	: Master Of Science
M/H	: Metro-Highway
Max	: Maximum
MIT	: Massachusetts Institute Of Technology
MOFT	: Minimum Oil Film Thickness
MPFI	: Multi Port Fuel Injection
MSI	: Motor Service International
NEDC	: New European Driving Cycle
NM	: Normal Mode
NNS	: Normal Normal Skip
NS	: Normal Skip
NS_N	: NS Skip Cycle Firing Part
NS_S	: NS Skip Cycle Not Firing Part
NSS	: Normal Skip Skip
NVH	: Noise Vibration And Harshness

OEM	: Original Equipment Manufacturer
PCJ	: Piston Cooling Jet
PCV	: Positive Crankcase Ventilation
PDA	: Port Deactivation
Ph.D	: Philosophy Of Doctorate
PMEP	: Pumping Mean Effective Pressure
PSA	: Peugeot Société Anonyme (Owner Of Peugeot And Citroen Brands)
PWM	: Pulse Width Modulation
RevBB	: Reverse Blow By
RPM	: Revolution Per Minute
SAE	: Society Of Automotive Engineers
SAPS	: Sulphated Ash And Phosphorus
SFC	: Specific Fuel Consumption
SI	: Spark Ignition
SOHC	: Single Overhead Cam
S-TDC	: Scavenging Top Dead Center
TAN	: Total Acid Number
TBN	: Total Base Number
TDC	: Top Dead Center
Temp	: Temperature
TO	: Throw Off
TS	: Thrust Side
VCT	: Variable Cam Timing
VOP	: Variable Oil Pump
vStroke	: Variable Stroke Volume Cylinder Deactivation
VTEC	: Variable Valve Timing And Lift Electronic Control
ZDDP	: Zinc Dialkydithiophosphate
η	: Dynamic Viscosity
π	: Pi Number

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THE EFFECTS OF CYLINDER DEACTIVATION ON ENGINE LUBE OIL CONSUMPTION

SUMMARY

Oil consumption has always been a major issue for internal combustion engines. Especially, for compression ignition engines due to higher combustion temperatures, longer stroke distances and higher loads oil consumption investigations have significant importance. However, spark ignition engines work faster in general.

Beside the environmental leak, there are different oil consumption mechanisms inside an engine. Engine oil sprayed to cylinder or liner walls from piston cooling jets directly participates in combustion and burns up. Another mechanism is the participation of oil vapour into combustion event through intake valves. Hot blow by gas mixture incorporates oil vapour and mist oil particles. This blow by gas mixture meets with intake air and then enters into combustion chamber. If the engine is equipped with a turbocharger, the oil leak from its bearing to compressor and turbine sides is the other oil consumption mechanism. Leaking to compressor side oil is carried inside cylinder by air flow. Oil leaking to turbine side creates pollution in the exhaust system.

Burnt in combustion chamber oil results in unwanted emission such as soot, sulphated ash or phosphated ash, which poison after treatment system, and so in violation of emission regulations. In addition to unwanted emissions, especially for spark ignition engines, dead end knock occurs due to oil existence inside the combustion chamber. On the other hand, automotive market demands longer oil service interval. Reduction of oil consumption leads to reduction in the degradation level of oil. Therefore, oil consumption investigations are important to satisfy market demand.

Beside the oil consumption investigations; increasing engine efficiency, due to the above explained reasons, is recently a hot topic in automotive industry. Especially for spark ignition engines cylinder deactivation is a good example of those efforts spent for efficiency increase. The aim of cylinder deactivation technology is decreasing intake air pressure loss through the throttle and so increasing the volumetric efficiency of the engine. This thesis would like to explain how the oil transfer mechanism changes between piston assembly and liner or cylinder wall if the working cylinder is deactivated.

This study focuses on oil consumption through cylinder or liner walls. Mainly, two different scenarios are compared on the basis of oil consumption. Those are regular combustion cylinder and deactivated cylinder. Also, in details of deactivated cylinder oil consumption calculations, skip cycle working strategy was questioned on the basis of oil consumption characteristics. In the study, single cylinder spark ignition engine oil consumption is calculated. AVL Excite program is used to model the scenarios. Thermodynamic input is gained from the program Ricardo Wave. The

effect of cylinder deactivation, which is a contemporary topic, on oil consumption through liner walls is clearly presented.

In these three different scenarios, the relationship among oil transportation through the piston assembly and the liner interface, piston and piston ring dynamics and thermodynamic cycle properties is investigated with the help of the AVL Excite model. During cylinder deactivation, whether it is a cycle skipping or permanently shutting the cylinder down, oil thrown off by the piston top edge and by the top ring upper surface increases due to thicker oil film present on cylinder wall when combustion does not occur. Also although the gas temperature in not fired cylinder is lower than the gas temperature in fired cylinder, amount of evaporated oil from liner surface inside to the cylinder is bigger in not fired condition due to vacuum pressure inside cylinder.

In general, the amount of oil transported by reverse blow by gas into the cylinder is much smaller comparing the other oil transportation mechanisms such as oil evaporation from liner surface and the oil scraped by piston top land edges and thrown off by top ring. The biggest contribution to oil consumption through piston and liner interface comes from oil evaporation. Those are valid for all working conditions. However, when firing does not occur, or in other words when cylinder is deactivated under vacuum pressure, oil transportation into combustion chamber increases.

SİLİNDİR DEAKTİVASYONUNUN İÇTEN YANMALI MOTORLARDA YAĞ TÜKETİMİNE ETKİLERİ

ÖZET

Yağ tüketimi içten yanmalı motorlar için her zaman sorun teşkil etmiştir. Özellikle yüksek yanma sıcaklıkları, daha uzun strok mesafeleri ve yüksek yükler nedeniyle dizel motorlar için yağ tüketimi araştırmaları daha da büyük önem taşımaktadır. Buna karşılık, benzinli motorlar ise daha yüksek hızda çalıştıklarından yağ tüketimi araştırmaları bu motorlarda yok sayılmamalıdır.

Motordan dışarıya sızan yağın yanında, motor içerisinde farklı yağ tüketimini mekanizmaları da vardır. Piston soğutma jetlerinden silindir veya gömlek duvarına püskürtülen motor yağı doğrudan yanmaya katılır. Başka bir mekanizma ise emme valflerini aşan yağ buharı ve yağ damlacıklarının yanma olayına katılımıdır. Silindir içi patlama sonrası piston sekmanlarını aşan sıcak gaz karışımı tarafından yağ buharı ve yağ partikülleri emme manifolduna kadar taşınır. Buradan da yanma odasına giren yağ buharı ve yağ damlacıkları yanarak tüketilmiş olur. Motor bir turbo ile donatılmış ise, turbo şaft yatağından kompresör ve türbin taraflarına yağ kaçağı olabilir. Bu da bir diğer yağ tüketimi mekanizmasıdır. Kompresör tarafına sızan yağ hava akımı ile silindir içine taşınır. Türbin tarafına sızan yağ ise egzoz sisteminde kirliliğe ve dolayısıyla fonksiyon bozukluklarına yol açar.

Yanma odasında yanan yağ, kurum, sülfatlı kül veya fosfatlı kül salınımına dönüşerek egzos sistemi komponentlerinin işlevsiz hale gelmesine sebep olabilir. Bu durum ise yasalarla düzenlenmiş emisyon standartlarının aşılması anlamını taşır. İstenmeyen emisyonlara ek olarak, özellikle benzinli motorlarda, yanma odasında yağ varlığı yanma esnasında vuruntu bölgeleri oluşturabilir. Vuruntu ise motor ömrü açısından son derece tehlikeli bir durumdur. Öte yandan, otomotiv pazarında yağ servis aralıklarının uzamasına yönelik bir eğilim vardır. Yağın bozulma hızının azalması yağ tüketiminin azalmasıyla doğru orantılıdır. Çünkü motorun servis aralığı süresince yağa katılan kurum, yakıt ve su gibi kirleticilerin oranı yağın miktarı azaldıkça artacaktır. Bu nedenle, yağ tüketimi araştırmaları pazar talebinin karşılanması açısından önemlidir.

Yağ tüketimi araştırmaları bir yana dursun, çevresel ve ekonomik kaygılardan ve düzenlemelerden ötürü özellikle yakıt tüketimini azaltmaya yönelik çalışmalar son zamanlarda hız kazanmıştır. Bu çalışmalarda önceliklendirilen konu motorun çalışma verimini artırmak yönünde olmaktadır. İşte bu amaçla da özellikle benzinli motorlarda uygulanan silindir deaktivasyonu teknolojisi dikkat çekmektedir. Silindir deaktivasyonu gaz kelebeğinin hava emiş hattında yarattığı basınç kaybını minimuma indirerek çalışan silindirlerin daha fazla hava ile beslenmesi sonucu motorun hacimsel verimini arttırmayı hedefler. Bu tez çalışmasında da deaktive

edilmiş silindirde piston komplesi üzerinden silindir içine yağ taşınması mekanizmasının ne şekilde değişebileceği açıklanmaya çalışılmıştır.

Bu çalışma, silindir ya da gömlek duvarından doğrudan yanma olayına karışan yağın miktarına odaklanmaktadır. Temel olarak iki farklı yağ tüketimi senaryosu bu tezde karşılaştırmalı olarak incelenmiştir. Bu senaryolar düzenli yanmanın olduğu bir silindiri ve deaktive edilmiş bir silindiri ihtiva eder. Deaktive edilmiş silindirlerin yağ tüketimi hesaplanırken çevrim atlama durumunda yağ tüketimi karakterinin de ne şekilde etkilendiği bu araştırmada yer almıştır. Günümüzde, benzinli motorlardaki emme kayıplarını azaltmaya yönelik geliştirilmiş silindir devre dışı bırakma mekanizması yaygınlık kazanmaktadır. Bu mekanizma silindiri tamamen kapatarak ya da silindire termodinamik çevrim atlatarak çalıştırılabilir. Çevrim atlatma yöntemi henüz akademik çalışmaların teorik bir konusu olarak literatürde yer almaktadır ve bu yöntemin ticarileşmesi için zaman gerekmektedir. Her ne kadar silindir pasifleştirme fikri benzinli motorlarda emme kayıplarının azaltılması şeklinde fayda sağlasa da ileriki zamanlarda dizel motorlarda da emisyon iyileştirme amaçlı karşımıza çıkabilir. Bu tezde tek silindirli benzinli motorun yağ tüketimi hesaplanmıştır. AVL Excite programı yukarıda belirtilmiş senaryoların modellemek için kullanılırken; termodinamik girdi Ricardo Wave programı ile elde edilmiştir. Gömlek duvarlarından direk olarak yanmaya katılan yağın miktarı çeşitli silindir deaktivasyonu durumları için kıyaslanmıştır.

Silindir deaktivasyonu senaryoları bir takım kabulleri içermektedir. Bu kabullere göre termodinamik data atmosferik emiş yapan 1.6L toplam hacme sahip 4 silindirli 4 zamanlı bir motorun tek silindirine aittir. Termodinamik data oldukça düşük yükleme durumu göz önünde bulundurularak üretilmiştir. Bunun sebebi ise normal çalışma şartlarına göre belirli bir yük eşiği üzerinde kalındığı taktirde, silindir deaktivasyonu yapıldığında aktif silindirler tam yükte çalışsa dahi motorun performansı normal çalışma durumundaki performansının altında kalma riskidir. Bir diğer önemli varsayım da senaryoların oluşturulması üzerine yapılmıştır. Termodinamik data 1440 derecelik krank açısı toplamına göre üretilmiştir. Normal çalışan, bir başka değişle 1440 derecenin karşılığı olan peşpeşe 2 termodinamik çevrimin ikisinde de yanmanın gerçekleştiği şartlara ait termodinamik data üretilmiş ve buna karşılık da yağ tüketimi araştırmasının temeli olan hesaplamalar yapılmıştır. Bu temel daha sonra hem çevrim atlama hem de değişken hacimli silindir deaktivasyonu senaryoları ile karşılaştırılmıştır. Bu kıyası yapabilmek için gerekli termodinamik data da yine peşpeşe iki termodinamik çevrim göz önünde bulundurularak üretilmiştir. Ama bu kez ilk çevrimde yanma gerçekleşirken ikinci çevrimde silindir içerisinde vakum olacak şekilde sübaplar kapatılmış ve yanmanın olmasına izin verilmemiştir. Çevrim atlama yöntemli silindir deaktivasyonu yağ tüketimi araştırmalarında ilk çevrimde yanmanın olduğu ama ikince çevrimde yanmanın gerçekleşmediği bu data kullanılırken değişken hacimli silindir deaktivasyonu yağ tüketimi araştırması yapılırken yanmanın gerçekleşmediği termodinamik çevrim tekrarlanıyormuş gibi varsayılmıştır. Ama unutulmamalıdır ki her iki termodinamik data da 1440 derece sonunda eşit iş yapıldığını göstermektedir. Bu eşitlik, silindir deaktivasyonu stratejisinin mantığının bir gereğidir.

Piston-silindir modeli kurulurken de birtakım varsayımlardan yararlanılmıştır. Örneğin çalışma şartları deaktivasyona göre değiştirildiğinde piston ve silindir sıcaklıkları sabit tutulmuştur. Böylece yağ taşınımında sadece termodinamik çevrimin etkisi gözlemlenmiştir. Sübap tarafından krank tarafına doğru aksiyal yönde hem piston hem de silindir için belirli bir sıcaklık gradyanı tanımlansa da radyal yönde sıcaklığın değişmediği varsayılmıştır. Krank boşluğu basıncı da atmosfer basıncını temsilen sabit olarak 1bar tanımlanmıştır.

Bu üç farklı çalışma şekli için, piston komplesi ve silindir duvarı arasından silindir içerisine taşınan yağ miktarı ile piston ve sekman dinamiği ve termodinamik çevrim özellikleri arasındaki ilişki araştırılmıştır. Araştırma yapılırken de krank mili dönüş hızı 2000rpm olarak programa tanımlanmıştır. Çalışmanın geneline bakılırsa, silindir içerine taşınan yağ miktarının normal çalışma şartları temel alındığında, çevrim atlama yöntemli senaryoda yaklaşık ikiye, değişken strok hacimli senaryoda ise yaklaşık üçe katlandığı görülür. Yağ taşınımındaki en büyük etkenin; çalışma şartlarından bağımsız olarak, yağın buharlaşması olduğu belirlenmiştir. Birinci sekmanın üst yüzeyinde biriken yağın atalet kuvvetleri altında silindir içine atılması da belirgin bir yağ taşınım mekanizmasıdır.

Çevrim atlamalı silindir deaktivasyonu ya da değişken stroke hacmi yöntemlerinden hangisi olursa olsun farketmeksizin, pistonun yanma odası tarafı kenarları ve birinci sekman tarafından atalet kuvvetleri ile silindir içerisine atılan yağ miktarı normal çalışma durumuna nazaran artmaktadır. Bunun sebebi silindir duvarında bulunan yağ filmi kalınlığının normal çalışma durumuna göre yanma olmayan silindirde artmış ve dolayısıyla da daha fazla yağın sekman yüzeyinde birikmiş olmasıdır.

Çalışmanın geneline bakıldığında da silindir içerisine olacak olası ters gaz akışının taşıdığı yağ miktarının neredeyse ihmal edilebilir düzeyde olduğu görülmüştür. Bununla birlikte yağ taşınımındaki en önemli etken yağın silindir duvarlarından buharlaşarak silindir içerisine taşınması olarak belirlenmiştir. Deaktivasyon esnasında yanma olmamasına rağmen buharlaşma hızının bu denli artıyor olması deaktivasyonun vakum altında yapılmasına bağlanmaktadır. Sonuçta ise silindirde yanma olmadığında ve silindir vakum basıncı altında deaktive edildiğinde, yanma odasına taşınan yağın miktarının arttığı anlaşılmıştır. Çünkü yağ filmi üzerindeki buhar basıncı düşmekte ve de silindir içinde yağ molekülü konsantrasyonu azalmaktadır. Yanma olmadığı zaman silindir yüzeyindeki yağ filmi kalınlıklarının arttığı da gözlemlenmiştir. Bu da buharlaşma potansiyeli olan yağ miktarının arttığı anlamına gelir. Yani silindir yüzezinde patlama olmadığı durumda daha fazla yağ bulunmaktadır. Bu durum sekmanların çapının patlama olmadığında daha küçük olması ile açıklanabilir. Çünkü patlama olmadığında silindir içindeki gaz sıcaklıkları da daha düşüktür. Ve ek olarak sekmanların düzgün çalışması için gerekli sekman arkası gaz basıncının patlama olmadığı durumda çok düşük seviyelerde kalması da unutulmamalıdır. Bu nedenle de özellikle birinci sekman yağı sıyırma görevini tam anlamıyla yerine getiremez.

Silindir içerisine taşınan yağ miktarında gözlemlenen artışın bu denli belirgin olması ve taşınımdaki ana mekanizmanın vakum kaynaklı buharlaşma olması, silindir deaktivasyonu çevriminin iyi araştırılması gerektiğine işaret etmektedir. Deaktivasyonun vakum altında değil de genişleme stroğunun akabinde sübapların kapatılmasıyla basınç altında yapılması buharlaşma ile taşınan yağ miktarındaki değişimin bu denli fazla olmayacağını düşündürmektedir. Öte yandan yapılan bu çalışmanın ileriki dönemlerde deneysel olarak da tekrarlanarak bulguların fiziksel olarak doğrulanması literatürü zenginleştirecektir.

1. INTRODUCTION

1.1 Motivation

Engine oil consumption is one of the primary research fields in the automotive industry due to tighter exhaust emission regulations and due to more demanding customer attitude. At this point it should be noted that internal combustion engine efficiency increase studies have always been continuing to satisfy those regulations and demands. In Figure 1.1, the dramatic evolution of passenger car diesel engine emission regulations is presented as an example. The detailed information that is summarized in the figure is presented in Table 1.1. The evolution at the emission regulations is of course not limited with diesel engines. The same trend is valid for spark ignition engines as well.

Current engine technology improves the thermal efficiencies up to 35% for gasoline engines and almost 50% for diesel engines at full load conditions [1]. However, especially for spark ignition engines, when running part loads, the efficiency drops down to 15% [1]. In order to increase part load efficiency, cylinder deactivation idea was generated. Cylinder deactivation can be achieved both by shutting off the engine or by skipping engine cycles. However, cycle skipping is yet a theoretical approach for efficiency increase. Until electronically controlled piston cooling jets are developed and introduced to market, excessive amount of oil is sprayed to cylinder wall which leads to yet unknown oil consumption behavior. And also oil consumption data gives the insight about some failure modes such as cylinder wear, bore polishing, piston ring dynamics, soot accumulation on piston surface and after treatment poisoning [2]. In this thesis, it is aimed to understand the oil consumption trend when cylinder is deactivated.

Although cylinder deactivation is currently a valid technology for gasoline engines to increase part load efficiency by reducing throttling loss for working cylinders or cycles, it can be a future application for diesel engines. In this study oil consumption is calculated on the basis of naturally aspirated gasoline engine thermodynamic data through a single cylinder oil consumption model. Regardless with the cylinder deactivation fundamentals, it is solely focused on oil consumption trends of deactivated cylinder.

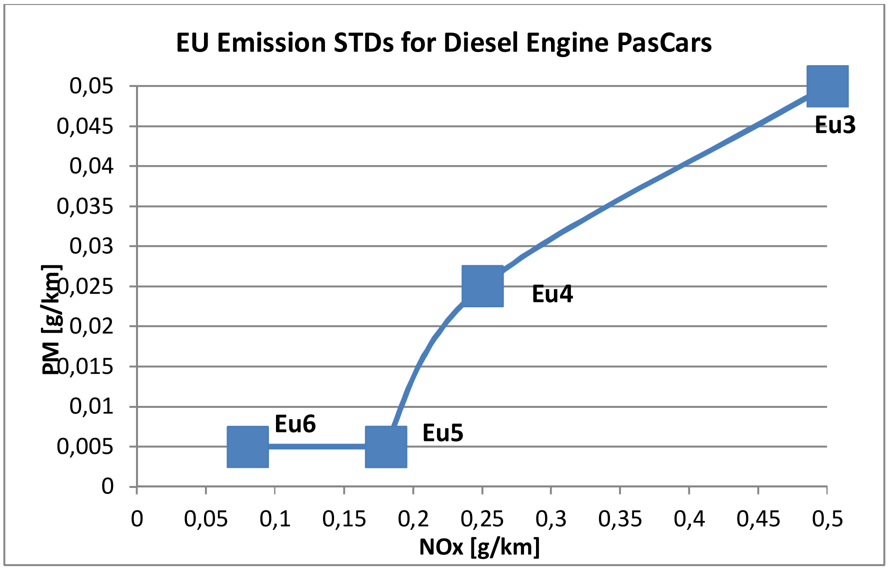


Figure 1.1 : EU emission standards for diesel engine passenger cars which shows the evolution by time [3].

Table 1.1 : EU emission standards for passenger cars (category M_1*) which shows the details of maximum allowable emission amounts [3].

Stage	Date	CO	HC	HC+NOx	NOx	PM	PN
		g/km					#/km
Compression Ignition (Diesel)							
Euro 1†	1992.07	2.72 (3.16)	-	0.97 (1.13)	-	0.14 (0.18)	-
Euro 2, IDI	1996.01	1	-	0.7	-	0.08	-
Euro 2, DI	1996.01	1	-	0.9	-	0.1	-
Euro 3	2000.01	0.64	-	0.56	0.5	0.05	-
Euro 4	2005.01	0.5	-	0.3	0.25	0.025	-
Euro 5a	2009.09	0.5	-	0.23	0.18	0.005	-
Euro 5b	2011.09	0.5	-	0.23	0.18	0.005	6.0×10^{11}
Euro 6	2014.09	0.5	-	0.17	0.08	0.005	6.0×10^{11}
Positive Ignition (Gasoline)							
Euro 1†	1992.07	2.72 (3.16)	-	0.97 (1.13)	-	-	-
Euro 2	1996.01	2.2	-	0.5	-	-	-
Euro 3	2000.01	2.3	0.2	-	0.15	-	-
Euro 4	2005.01	1	0.1	-	0.08	-	-
Euro 5	2009.09	1	0.1	-	0.06	0.005	-
Euro 6	2014.09	1	0.1	-	0.06	0.005	6.0×10^{11}

* At the Euro 1..4 stages, passenger vehicles > 2,500 kg were type approved as Category N1 vehicles

† Values in brackets are conformity of production (COP) limits

1.2 Background

1.2.1 Lubrication system overview

Lubrication system has a vital role in an internal combustion engine. There are basically two different groups of engine lubricated components. One of the groups includes the components which require oil flow. All kinds of bearings are quite good examples of this group. The other group includes the engine components which require oil pressure such as piston cooling jets, hydraulic lash adjusters and variable cam timing mechanisms in valve train system. Therefore, beside the tribological and lubrication features of the oil, it has crucial effect on functionality of the engine components. Please see Table 1.2 for detailed relationship between engine lube oil and the other components.

Although ‘lubrication’ has a term meaning that wear and friction reduction by introducing a lubricant between moving surfaces in proximity, lubrication system has many functions in engine. They are [2], [4]:

- Increasing mechanical efficiency by frictional resistance reduction.
- Protecting engine against wear.
- Cooling effect on friction surfaces by energy dissipation.
- Piston cooling by direct spray.
- Protecting engine against scratch by removing injuring impurities.
- Keeping gas and oil exchange between piston and cylinder or liner walls at an acceptable minimum level.
- Noise reduction and damping of impact and vibration.
- Corrosion protection.
- Cleaning.

Table 1.2 : Lube oil functions for engine components [5]. For some components oil pressure and for others oil flow is required.

Engine Component	Oil Feed Type	Oil Function For The Component
Main bearings	Pressure feed	Need flow to replenish the oil film and for some cooling effect. Need pressure in its oil groove to feed oil to conrod bearings, which increases with engine speed. End up leaking much more than it is needed
Connecting rod big end bearings	Pressure feed Low requirement	Need flow to replenish the oil film and for some cooling effect.
Connecting rod small end bearings	Splash feed	Need flow to replenish the oil film and for minor cooling effect.
Piston cooling jets	Pressure feed	Flow requirement, impingement velocity may lead to pressure requirement.
Cam shaft bearings	Pressure feed Low requirement	Need flow to replenish the oil film and for minor cooling effect.
Variable cam timing system	Pressure feed	Hydraulic type needs pressure to unlock the pin and move it faster enough. Need sufficient flow capacity to keep up with the 'gulping' of the chambers. Cam torque type needs pressure to work through valves and some flow capacity to fill the chambers.
Chain drive system	Splash feed / Pressure feed	Chain tensioner needs pressure to work through the check valve and replenish the pressure chamber. Chain squirter needs flow, but can be sized to work with a given pressure.
Valvetrain system	Pressure feed	Hydraulic lash adjusters need pressure to work through the check valve and replenish the high pressure chamber. Supply sufficient oil through the squirter hole for others.
Turbocharger	Pressure feed	Has a pressure requirement to get sufficient oil flow to the system / bearings.
Vacuum pump	Pressure feed	Need oil flow to lubricate the system.
Fuel pump drive	Pressure feed	Need oil flow to lubricate the system.

Engine lubrication system is driven by an oil pump. There are several types of oil pumps. In conventional engines, there are mainly three types of oil pumps occupied. They are gear pumps, crescent pumps and rotor pumps.

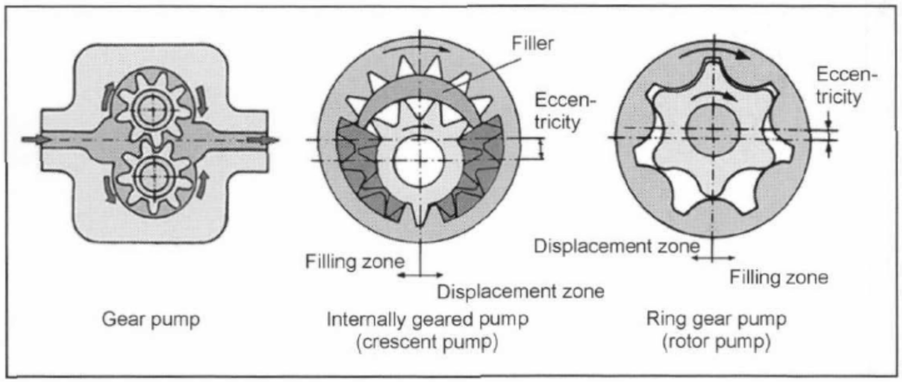


Figure 1.2 : Conventional oil pump types [2]. External gear pump, internal gear pump and gerotor pump are shown from left to right.

However, as the emission legislations and fuel economy demand of customers force automotive engineers to develop variable technologies. With this trend, variable flow oil pumps are nowadays working on passenger car engines. When engine is not loaded or is working at lower speeds, lubrication system demands less oil flow than it demands at high load conditions. This difference at the oil flow demand is perceived as hydraulic energy saving potential.

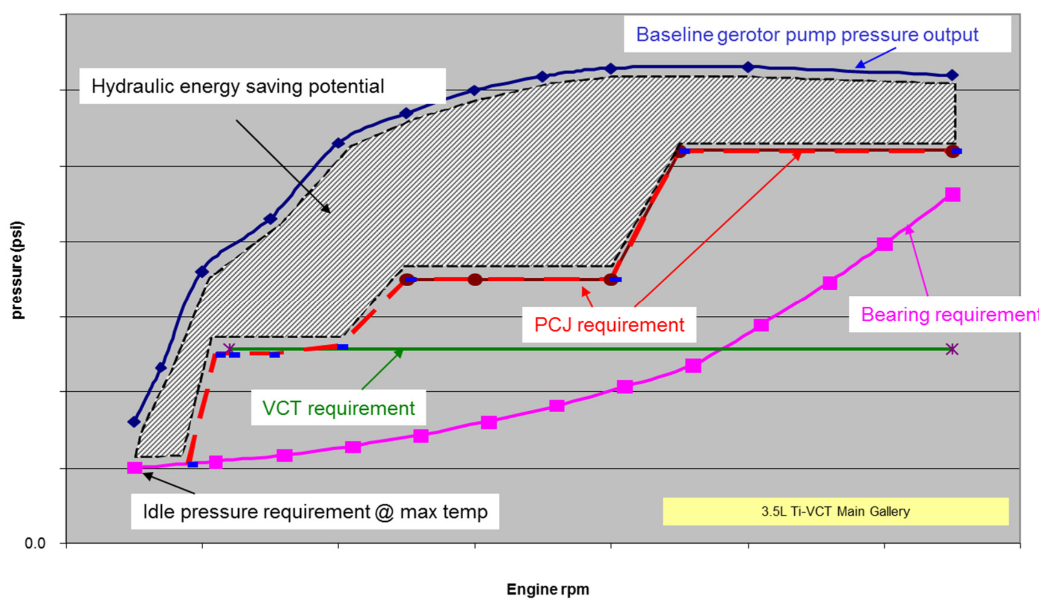


Figure 1.3 : Hydraulic energy saving potential of an internal combustion engine and variable flow oil pump development opportunity [5].

There are again three types of variable flow oil pumps. They are passive control VOP, two stage VOP and fully electrical controlled VOP. In passive control VOP technology, oil flow is adjusted by the main oil gallery pressure. In the other two VOP technologies oil flow is again adjusted according to the feedback coming from main oil gallery; however, this feedback is controlled by a valve. For two stage VOP an on/off valve is used while for fully electrical control VOP a proportional valve is used. The proportional valve is controlled by PWM signals. Therefore, the volumetric displacement of an electrical controlled VOP can be changed continuously.

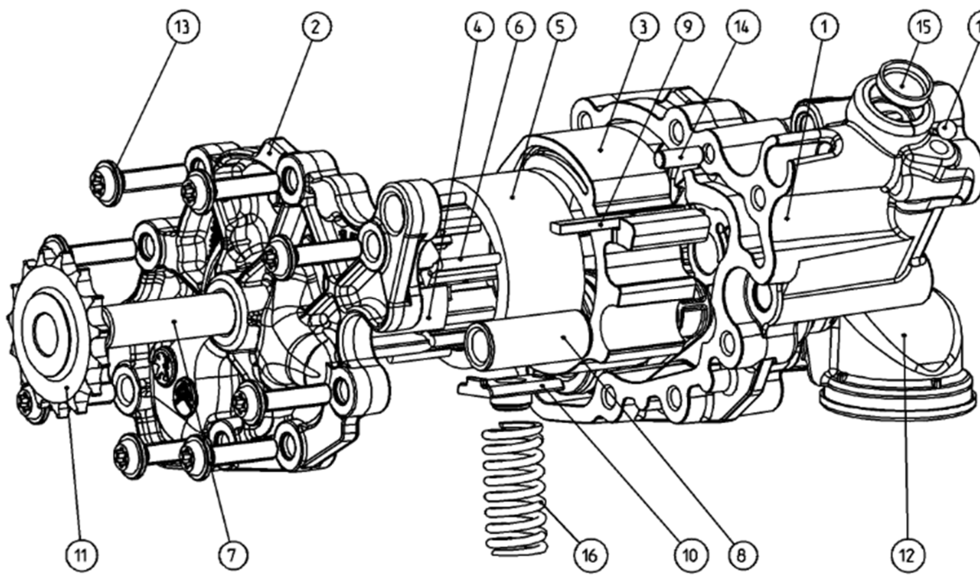


Figure 1.4 : Exploded view of VOP. 1- pump housing, 2- pump cover, 3- control slider, 4- inner rotor, 5- outer rotor, 6- pendulum, 7- shaft, 8- bearing tube, 9- sealing strip, 10- spring support, 11- sprocket, 12- oil suction pipe assembly, 13- bolts, 14- dowel pin, 15- seal cap, 16- regulation spring, 17- ball [5]

Oil pump sucks the oil inside the oil pan by the help of suction pipe reaching to the bottom of the pan and pushes inside to engine. Oil pan most often acts as a sump and keeps all engine oil inside itself beneath crankshaft. However, especially in high performance automobile engines, dry sump technology which is located separate than the engine is applied [6]. Sometimes, this concept can be a good solution for tight package problems. Dry sump system requires a second oil pump which is used to immediately evacuate the returning oil from the oil pan and send it to a separate reservoir from where it is again supplied to the engine. The resulting crankcase vacuum reduces pumping losses in the crankcase associated with piston movement. It also reduces the oil mist and further losses associated with fluid friction between the crankshaft assembly and the air and oil vapor in the crankcase.

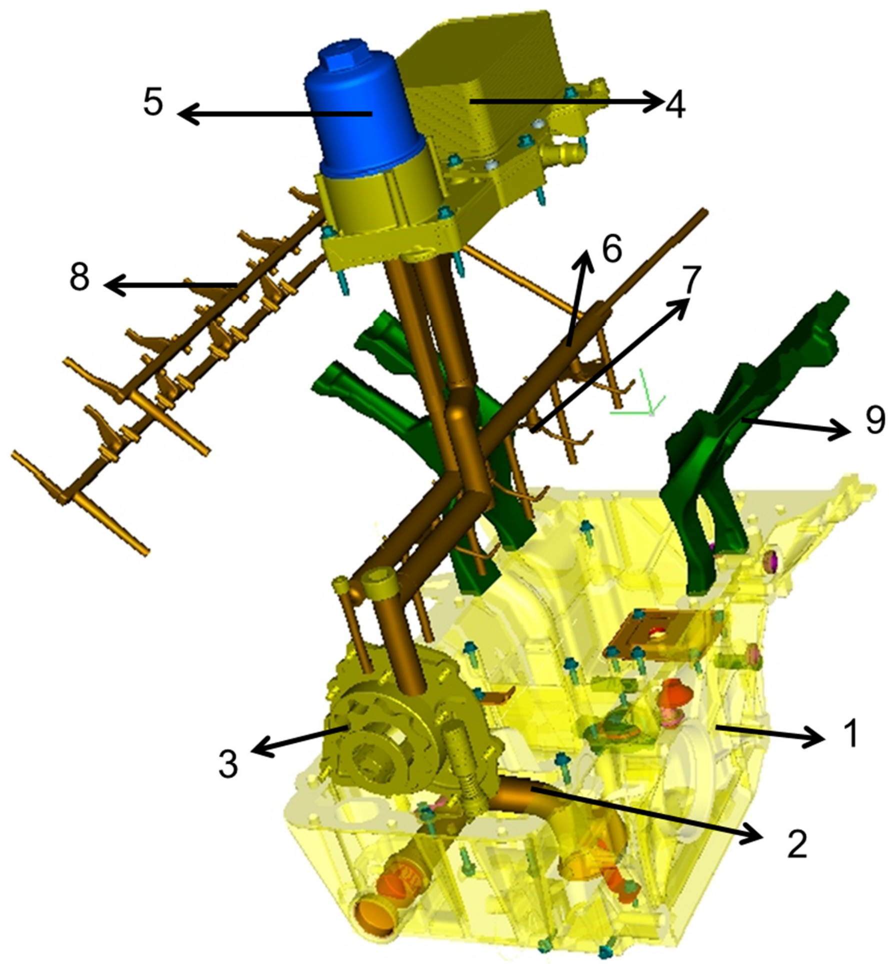


Figure 1.5 : V8 engine lubrication system schematic. 1- structural oil sump, 2- suction pipe, 3- rotor type oil pump, 4- oil cooler, 5- oil filter, 6- main oil gallery, 7- PCJ, 8- cam oil gallery, 9- oil return channels [5].

Especially in diesel engines as the metal temperatures are higher than that of in gasoline engines, an oil cooler is located in the lubrication circuit. If it is required, an oil cooler can also be a part of gasoline engines as well. Just before oil enters to the engine and is distributed through the engine, it is cooled down by engine coolant flow while passing the oil cooler. And then oil is filtered. Oil filter is always located after oil cooler in the lubrication circuit to protect oil against any potential corrosion problems inside the oil cooler as it is vulnerable against corrosion due to engine coolant flow. In general oil cooler and oil filter are modulated in an oil filter and cooler assembly.

The most important component of lubrication system is the lubricant, in other words the engine oil. Lubricant technology development sits on the basis of some principals such as customer demands and emission regulations. In general, for customer point of view, cost of ownership is quite important. This means long oil drain intervals make the engine preferable. To achieve longer service intervals, oil degradation rate should be slow. Oil should include dispersants, detergents, over base additives called TBN to neutralize acid accumulation quantified by TAN, anti-wear and anti-corrosion additives, viscosity index improvers and anti-oxidants [6]. The oil quality is strongly dependent on those additives. Most recently, automotive OEMs apply oil quality monitoring algorithms inside the ECU to allow flexible oil drain intervals. Those algorithms guess oil condition by following driver movements.

For the emissions point of view, ash quantity in exhaust gas due to combusted oil has significant influence. This is why engine oils are also classified as low SAPS, mid SAPS and high SAPS. The focus of this thesis work, oil consumption results in ash formation after combustion. Because, the oil amount transferred into combustion chamber through piston and liner interface is discussed. High amount of ash blocks DPF quickly. Also sulphated ash poisons LNT and makes it out of function. Therefore, market demand tents to low SAPS level lubricants as the emission legislations get tighter.

1.2.2 Oil consumption mechanisms

The oil level in the oil sump decreases by time due to oil losses and oil consumption. Oil losses occur when oil escapes through the rigid and moving joints of the engine. These can be the connection from the crankcase to the sump and cylinder head, the connection from the cylinder head to the cylinder head cover, the connections between oil filter and oil cooler, as well as leaking oil drain plugs and crankshaft seals. The actual oil consumption results from internal leaks resulting in burning and evaporation of oil. Such leaks are mainly caused by worn piston rings or piston ring grooves, excessive clearance between valve stem and valve guide or leaks in the turbocharger. The oil consumption can be estimated only roughly because it depends on a large number of parameters which change during operation of engine [2].

Maximum acceptable oil consumption for a car engine can be up to 0.1L per 1000km [5]. Former explained internal leaks are visualized in the schematic in Figure 1.7.

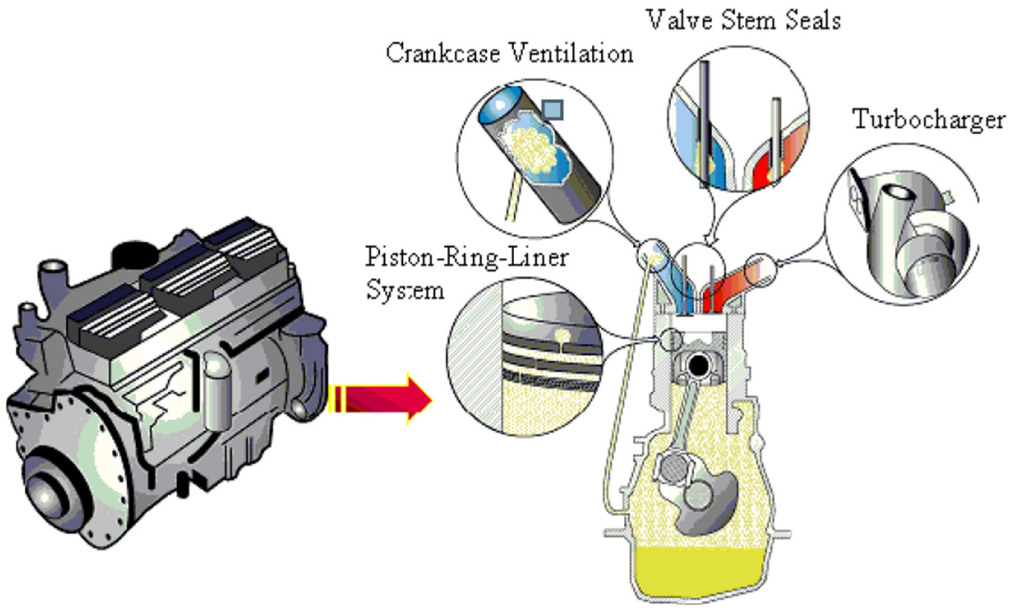


Figure 1.6 : Internal oil leak locations inside the engine which leads to oil consumption [7].

In general, oil consumption in an engine can be quantified as 0.001 times the fuel consumption value [5]. This proportion can give a rough estimation and an idea about the oil consumption values.

Table 1.3 : Oil consumption and oil loss sources in an engine by percentages [8]. The numbers were adapted from MSI failure statistics.

The Source of Oil Consumption (%)		The Source of Oil Loss (%)	
Misassembly	12	Component interface failures	15
Turbocharger	16	Sealing failures	14
Improper combustion	16	Contamination	14
Fuel injestion pump	24	Radial leakage	41
Irregular service	26	Others	16
Others	6		

Turbochargers wheels rotate with high rotational speeds such as 200000 rpm to pump high amount of air inside the combustion chamber for high level of cylinder pressures [5]. Turbocharger bearing is continuously lubricated by pressurized oil for wear protection and bearing cooling. The lubrication regime is hydrodynamic lubrication and the oil lubricating the bearing is evacuated by a turbo oil drain pipe back to crank case and to oil sump. While there is a positive exhaust gas pressure on turbine side of turbochargers, compressor sucks the air first and then pumps to intake manifold. This suction creates a vacuum on compressor wheel. Turbocharger bearing is sealed by small components which are called piston rings. The piston rings are located at both turbine and compressor sides of the bearing. The positive gas pressure on turbine helps turbine piston ring for a better sealing performance; however, the vacuum on compressor side has a negative effect on compressor piston ring. Also by time, due to severe working temperatures and high rotational speeds, turbocharger bearing clearances increase. Therefore, piston ring sealing performance reduces. Oil starts leaking first to compressor due to vacuum effect and then to turbine. Turbine leaking oil leads to after treatment system blockage. Oil leaking to compressor results in two different failure modes. One is, as mentioned before, oil consumption in the combustion chamber and the other failure mode is compressor coking. Compressor coking is a phenomenon which leads to engine performance reduction and compressor breaking at late stages. Ruptured compressor particles may lead to bigger damages on pistons, cylinder head or valves [8].

Up to now, expected oil leak mechanism inside a turbocharger due to wear by time, to turbo bearing seal efficiency and to turbocharger working nature is explained. However, sometimes unexpected oil leakage may happen from bearing to compressor and turbine if turbo oil drain pipe is blocked partially or fully. In this case oil feed flow is bigger than oil evacuation flow through the bearing. Therefore, oil pressure inside bearing increases and turbo shaft piston rings cannot resist any more [8].

a) b)

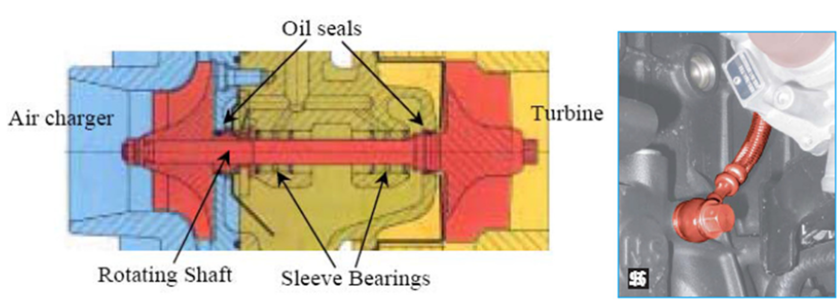


Figure 1.7 : a) Turbocharger schematic showing compressor wheel at left hand side, turbine wheel at right hand side, shaft connecting turbine wheel to compressor wheel, shaft bearings and piston rings as oil seals. [7]. b) Turbo oil drain pipe and connection of the pipe to the engine block [8].

Similar oil consumption mechanism is valid for fuel injection pump and also for valve stem seals. Again increased clearances due to wear may lead to oil leakage inside FIP and through valve stem seals. Oil leaks inside fuel line when lubricating FIP. Leaking inside fuel line oil is injected inside cylinders with fuel. Oil fills cylinder head cavities after lubricating cam shafts. And then it drains down to oil sump by gravity. However, especially when the air intake pressure gets below atmospheric pressure, oil can be transferred from cylinder head cavities through valve stem seals due to the pressure gradient. This leaking oil also joins to combustible mixture [8].

Oil can also be carried inside blow by gas into the cylinders. 'Blow by gas' term is used to describe exhaust gas leaking inside crank case. Leak path consists of piston to cylinder or liner interface, piston rings, valve stem seals, vacuum pump, and turbo oil drain pipe. Accumulation of blow by gas inside the crank case cannot be allowed as increasing gas pressure blows engine sealing components and also creates a restriction for piston reciprocating movement. Therefore, crank case ventilation system functions to evacuate blow by gas from crank case. Crank case ventilation can be done by either positive crank case ventilation system or closed crank case ventilation system. In PCV system, blow by gas is released to atmosphere through the pressure gradient from crank case to outside the engine. The pressure gradient occurs because accumulated blow by gas pressure is higher than atmospheric pressure. Recently, PCV systems can rarely be seen as the environmental regulations are getting tighter. CCV systems are replacing PCV systems. In CCV system, crank case is connected to intake manifold. Therefore, blow by gas contributes to combustion as similar as EGR system does. But the difference is because blow by gas passes over oil sump. In crank case due to high temperature and high crank shaft rotational speed oil vapor and oil mist exist. The higher oil level is the bigger amount of oil mist is in the crank case. Those oil vapor and very small oil droplets are carried into cylinders by blow by gas. Although oil separation function is implemented in the CCV system it is not possible to make all oil existence inside the blow by gas be diminished [8].

a) b) c) d)

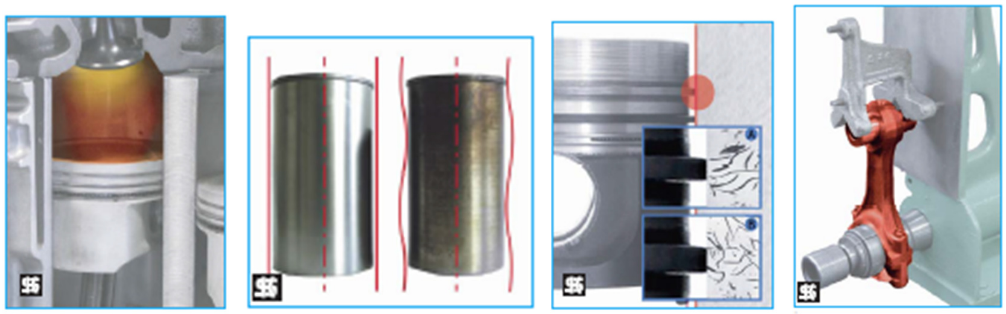


Figure 1.8 : a) Crossectional view of a cylinder, b) Bore distorsion visual aid, c) Graphite structure inside cylinder walls and piston, piston rings and cylinder wall interface, d) Connecting rod control gauge [8].

The most important oil consumption mechanism happens between piston and cylinder wall. As long as oil exists between piston and cylinder, some amount of this oil is consumed during combustion. This is not a failure mode, but is just the nature of an internal combustion engine. Oil consumption and friction reduction researches generally focus on this location. Increasing clearance always leads to increase in oil transportation. This is also valid for piston – cylinder interface. Injected fuel spray may reach to cylinder wall and dilutes the lubricating oil. Due to the dilution, oil viscosity reduces and the lubrication regime between piston and cylinder is adversely affected. This phenomenon causes cylinder wear. Also distorted by combustion gas pressure bore defines the functionality of piston rings as it also effects piston and piston ring dynamics. Improper geometry connecting rods also disturb the dynamics of piston and so the piston rings. Bore honing parameters also have significant influences on oil consumption. Honing channels and graphite vanes on the surface of bore holds the oil to create protective lubricating regime. However, evaporation and being swept of this oil layer cause oil consumption inside cylinder as a part of combustion [8].

Next section explains the details of the oil consumption mechanism between piston and bore.

1.2.3 Oil transportation through piston and cylinder interface

As the focus of this thesis work is understanding oil consumption through the interface between piston assembly and cylinder, oil transportation over piston rings has great importance for this study. The company AVL explains the oil transport mechanisms in its Excite Piston And Rings Module user manual in 4 different categories [9]:

- Evaporation at the oil film from the liner surface.
- Oil transportation around the first piston ring, including oil throw off.
- Oil blow through the gap of the first ring from and into the combustion chamber.
- Oil scraping at the top land's top edge considering deposits.

The M.Sc. thesis written by Senzer in 2007 in MIT includes similar explanations with the above listed mechanisms for oil transportation over piston rings. Senzer investigates the effects of piston ring pack design on spark ignition engine oil consumption [10].

It is suggested in this study that especially during low load working conditions due to bore distorsions oil can accumulate over piston crown and the upper flank of the top ring. The accumulated oil is then thrown into the combustion chamber either by piston crown edge or by the top ring. AVL names these two actions as oil throw off by first ring and oil scraping by the top land's top edge. Basically, the reason for both the throw off and the oil scraping is the inertial forces acting on accumulated oil. Therefore, it can be concluded that oil throw off and oil scraping occur at higher speed as the inertial forces increase with speed.

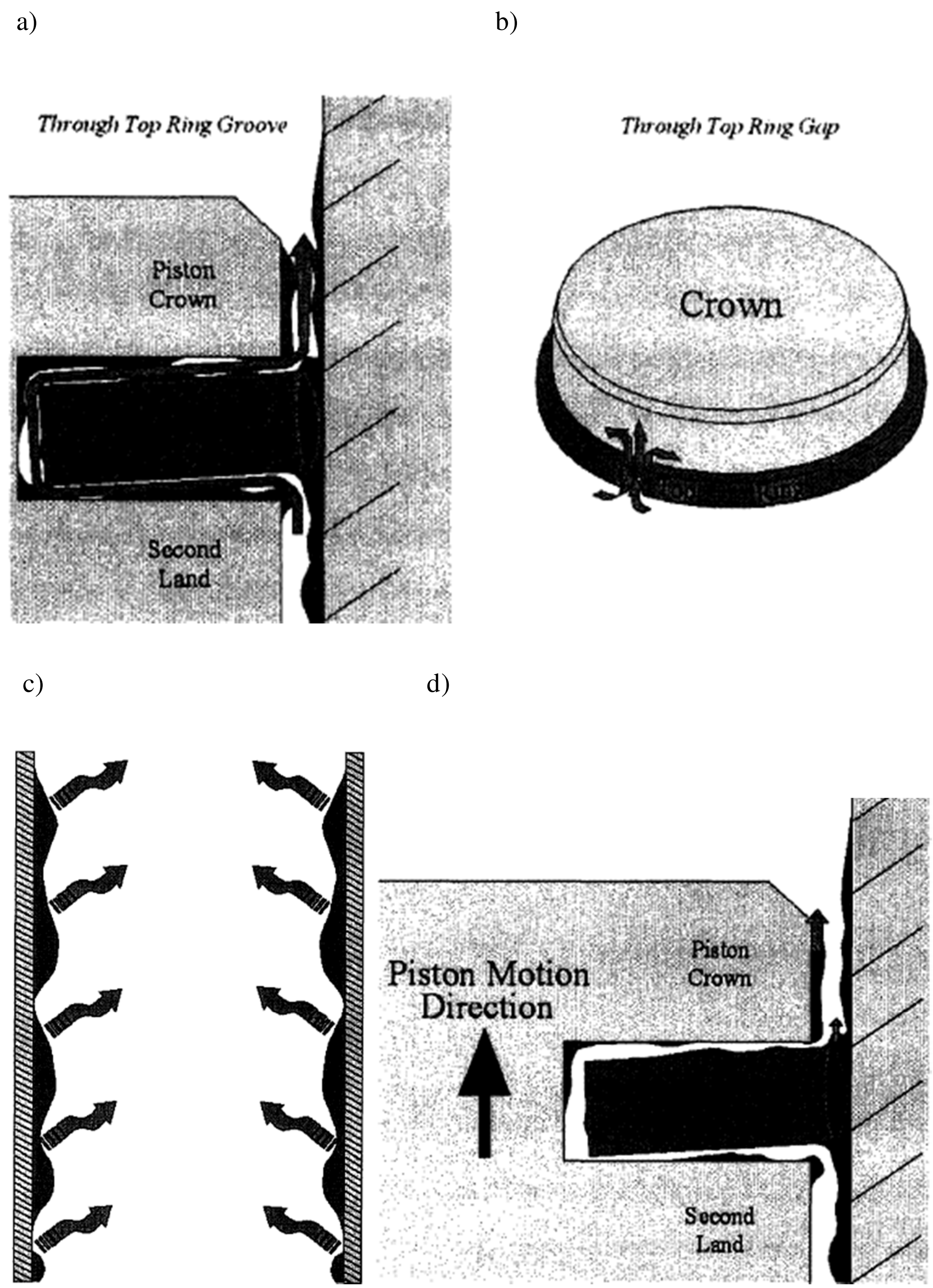


Figure 1.9 : a) Oil transport by reverse gas flow and oil pump up into combustion chamber by top ring, b) Oil transport by reverse gas flow through top ring close gap, c) Oil evaporation from liner surface to cylinder and to the combustion chamber, d) Oil throw off by inertial forces [10].

Especially during high load running condition, as the gas temperature raises, oil evaporation increases. The previous studies showed that the main contributor of total oil consumption in an engine is the oil evaporation. Of course the thermal conditions of the engine are determinant for the oil evaporation ; nevertheless, liner surface finish has also effects on oil evaporation. Liner surface profile can be thought as mountains and valleys where oil is present in the valleys. Consequently, the depth of the valleys is indicative for the oil amount present on the liner surface. Also oil composition is important for oil evaporation ; because, its chemical decomposition results in how volatile the oil is [10].

All the above explained oil transport mechanisms are very well schematized by Yilmaz (2003) below in Figure 1.11 [11].

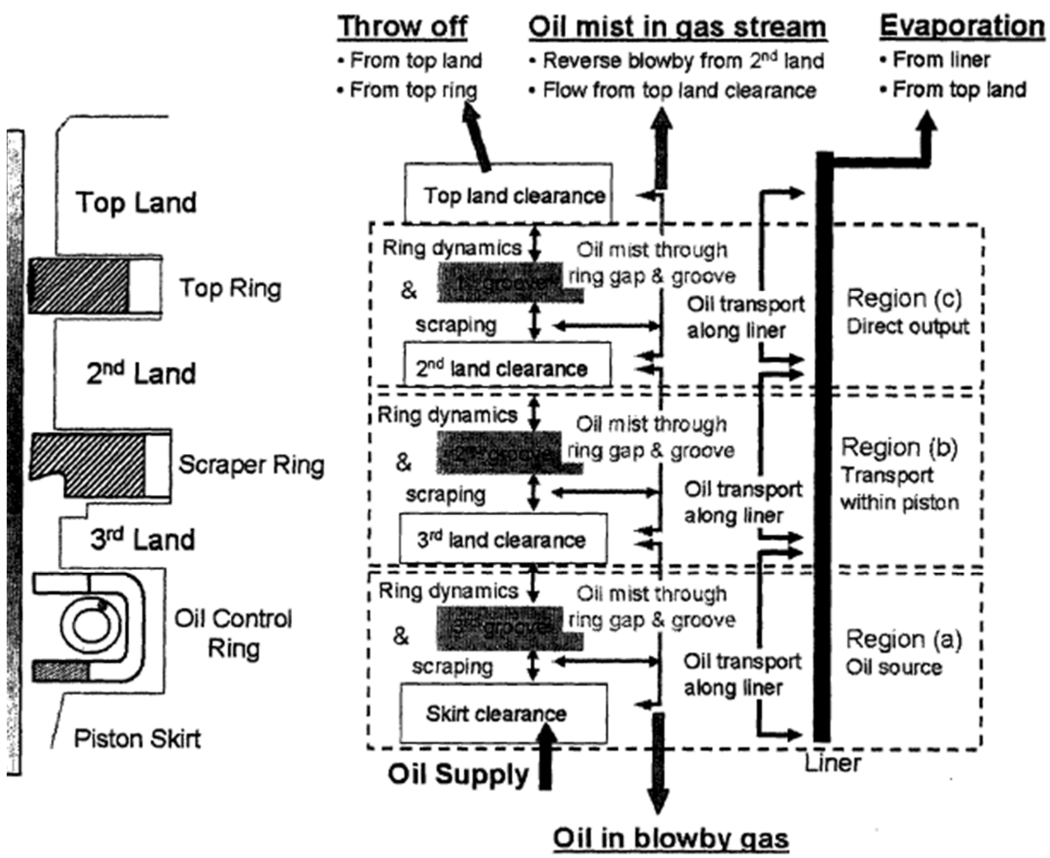


Figure 1.10 : Schematic of oil transport through the piston-ring-liner system clearances [11].

Referring Figure 1.10, region (a) includes the clearances in the upper piston skirt and the oil control ring groove. The sprayed by piston cooling jet oil has to pass region (a) to reach the second and top piston rings. In other words, region (a) controls the oil amount covering cylinder liner and transfered to upper piston regions [11].

Region (b) includes the clearances in the third land and the second ring groove. The same situation which was explained for region (a) is also valid for region (b). Oil has to pass region (b) to reach upper piston regions. Therefore, this region is determinant for the oil supply to second land and to the top ring groove [11].

Region (c) includes the clearances on the second land and top ring groove. This region is the first degree neighbour of the combustion chamber. Therefore, this region determines the contribution of direct oil consumption sources such as oil evaporation, thrown off oil or scraped oil into the combustion chamber [11].

As a summary, engine load and engine speed are the parameters which have significant influence on oil consumption. Engine load basically determines the cylinder pressure and temperatures. Therefore oil evaporation from the liner surface into the combustion chamber is mainly affected by engine load.

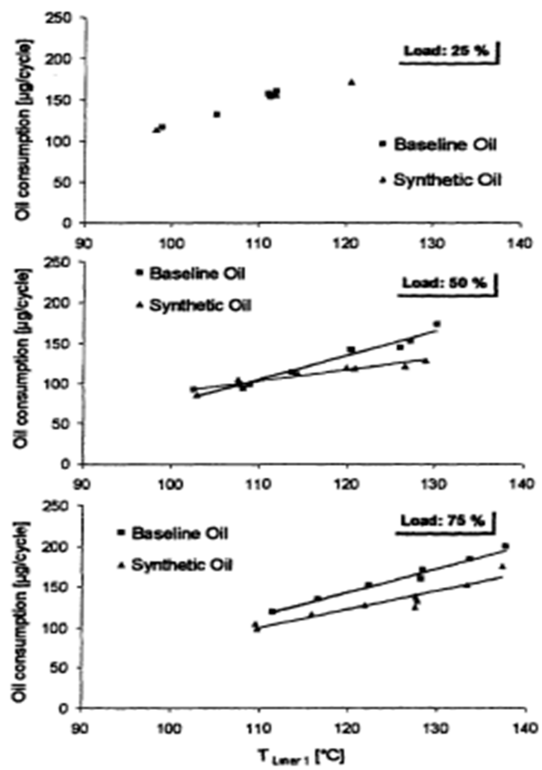


Figure 1.11 : Oil consumption dependence on liner temperature at different load and constant speed of 3500rpm [11].

Yilmaz explains this phenomena with some test results shown in Figure 1.11. As the load increases keeping the engine speed constant, liner temperature increases and so the oil evaporation does. In that study, the effect of oil composition on oil evaporation is also clearly detected. As the volatility of synthetic oil is lower than a baseline oils volatility, oil evaporation rate is smaller for synthetic oil.

Together with engine load, engine speed determines the piston and rings dynamic movements. Because, the gas pressure distribution over the piston rings is a function of load and speed. Therefore, transported over rings oil amount is basically affected by engine speed and engine load.

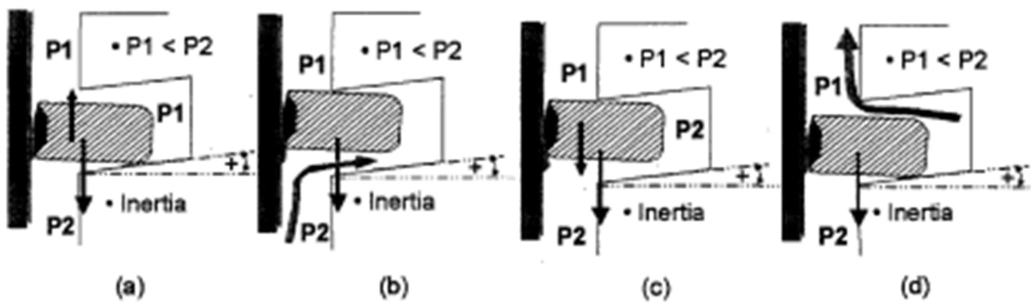


Figure 1.12 : Illustration of the sequence of events during top ring reverse flutter [11].

Referring Figure 1.12, during expansion stroke top ring sits at the bottom of its groove as the cylinder pressure is bigger than the second land gas pressure. However, at the end of the expansion stroke inter-ring pressure becomes higher than the cylinder pressure. Therefore, due to this pressure difference, top ring is lifted in its groove and the gas present in the second land penetrates into the top ring groove. Then increased ring back pressure helps top ring to achieve sealing function. Finally inertia force dominates the static equilibrium and top ring sits again at the bottom of its groove. The gas squezed between top ring and the piston then is released into the combustion chamber [11].

If the above explained working mechanism does not function properly for the top ring motions due to inequilibrium of the forces acting on top ring, top ring flutters. Flutter phenomena actually is caused by improper distribution of gas pressure beginning from combustion volume and through the piston cylinder interface. Top ring flutter impacts thrown off oil amount into the combustion chamber.

1.2.4 Cylinder deactivation strategy

Cylinder deactivation is basically an efficiency increase strategy especially for spark ignition engines. Although nowadays spark ignition engine efficiencies are increased up to around 40%, in part load conditions due to the restriction in air intake path created by throttle valve efficiency may decrease down to around 15% [1]. The throttle restriction creates a pumping loss through the air intake path. Knowing that, different kind of applications such as variable valve timing, stratified charge and direct injection, turbocharging or supercharging are developed. The most recent development among engine technology to increase the efficiency is the variable stroke volume application. This application is introduced to market by limited automotive OEMs such as Chrysler and Volkswagen.

Another concept of cylinder deactivation is 'skip cycle method'. Skip cycle method is a kind of partial deactivation of cylinder. This concept is now a theory which is being studied by universities. It suggests gaining work after each 8 or 12 strokes of engine.

The main purpose of cylinder deactivation is supplying torque demand at part loads with less number of working cylinders or less number of combustion cycles. To achieve torque demand with those explained constraints working cylinder or successful combustion pressures are increased. In other words, required work is supplied by limited number of working cylinders or the work gained after 2 conventional combustion cycles is equal to the work gained after 1 combustion and 1 skipped cycle for example. This means that during suction stroke less pressure drop occurs at the inlet of cylinder compared to conventional working mode and more air goes into the cylinder. Therefore, pumping loss across the throttle is reduced [1].

In diesel engines the combustion is controlled by the fuel injection quantity only. Therefore, none of those improvements are applicable for diesel engines except turbocharging and / or supercharging. However, in order to control diesel emissions variable valve timing and cylinder deactivation may be future development actions.

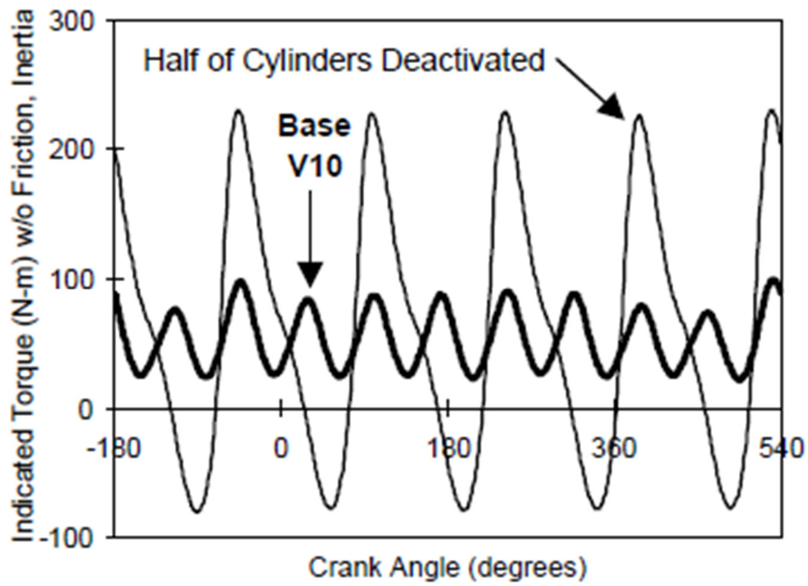


Figure 1.13 : Torque pulsation with cylinder deactivation. The average torque is equal for both normal and cylinder deactivated modes to supply required work which is kept as same for both situations [12].

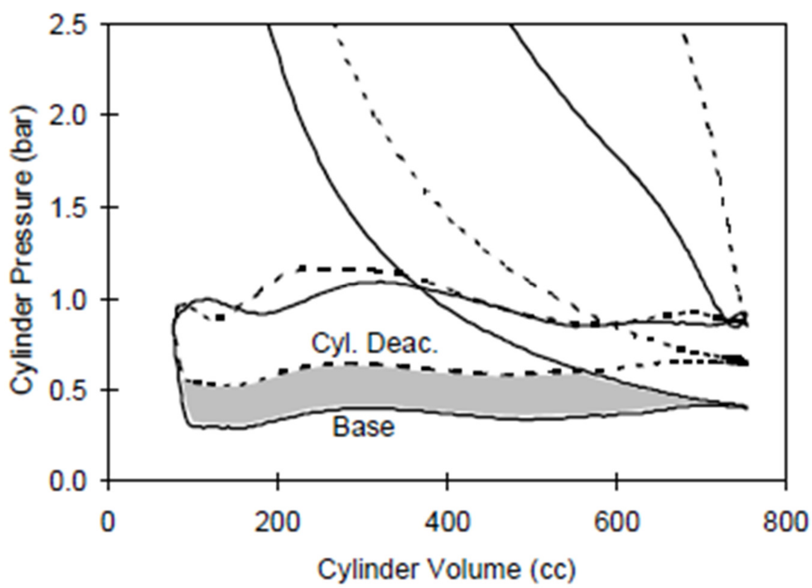


Figure 1.14 : Shaded area in the above graph shows the pumping work benefit of cylinder deactivation mode over the conventional working mode [12].

The above two figures show the difference between the two operation modes for the engine clearly. As the work demand does not change for one operation point of the vehicle, when the cylinders are deactivated, the torque response shows a wider torque fluctuation region. Nevertheless, the average torque stays same because the work demand does not change. As the torque amplitude increases the pumping loss decreases and results in efficiency increase.

1.2.4.1 Variable stroke volume

Variable stroke volume is the efficiency increase method which proposes reducing active working cylinder quantity in the low load region of the engine. In past, General Motors applied this method in their V8-6-4 engine for Cadillac. However at that time engine control technology was not developed enough to manage this method in the engine properly. Therefore, this engine vanished in 1981. This engine was a V8 structured engine but was capable to run with 6 or 4 active cylinders also. GM claimed 30% fuel consumption reduction with this application [1].

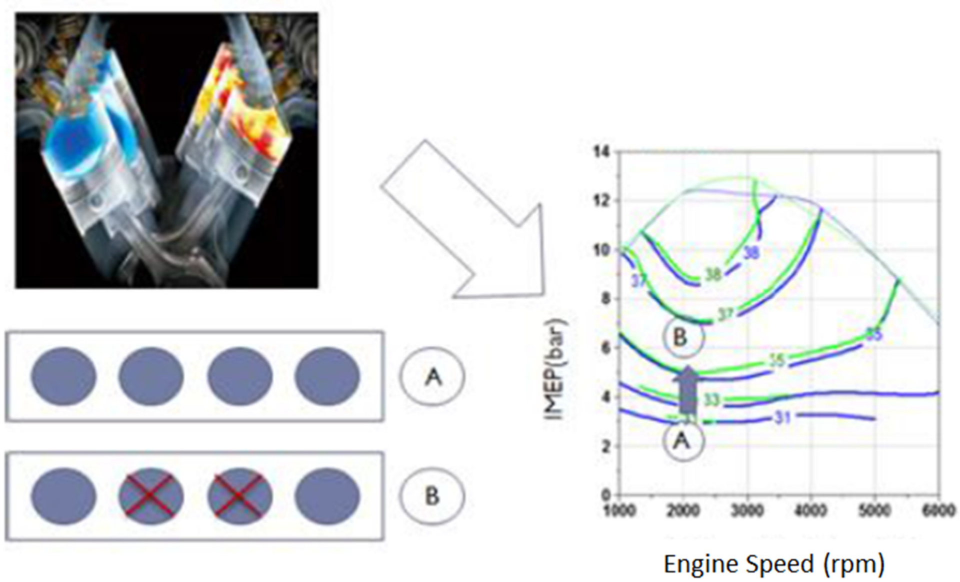


Figure 1.15 : When the middle 2 cylinders are deactivated in situation B cylinder pressure point shifts from A to B in the map at right hand side [1].

Figure 1.15 is a good example to understand how the active cylinder pressures are shifted if some of the cylinders deactivated. As the total torque demand does not change the required work is done by fewer cylinders compared to conventional working mode. The only way to supply the torque demand with fewer active cylinders is gaining more torque from those active cylinders. Consequently, active cylinder pressure increases.

When cylinders are deactivated, deactivated cylinder valves are decoupled from the camshaft motion in order to create a totally closed media not to lose pumping work on atmospheric air. The only loss would be due to the friction during piston motion inside the cylinder. DaimlerChrysler achieved this decoupling action in their 5.7L V8 HEMI engine by a lifter mechanism implemented on the pushrod. The lifter looks like a hydraulic lash adjuster (HLA) and acts as if it is an HLA during normal operation mode [13].

a) b)

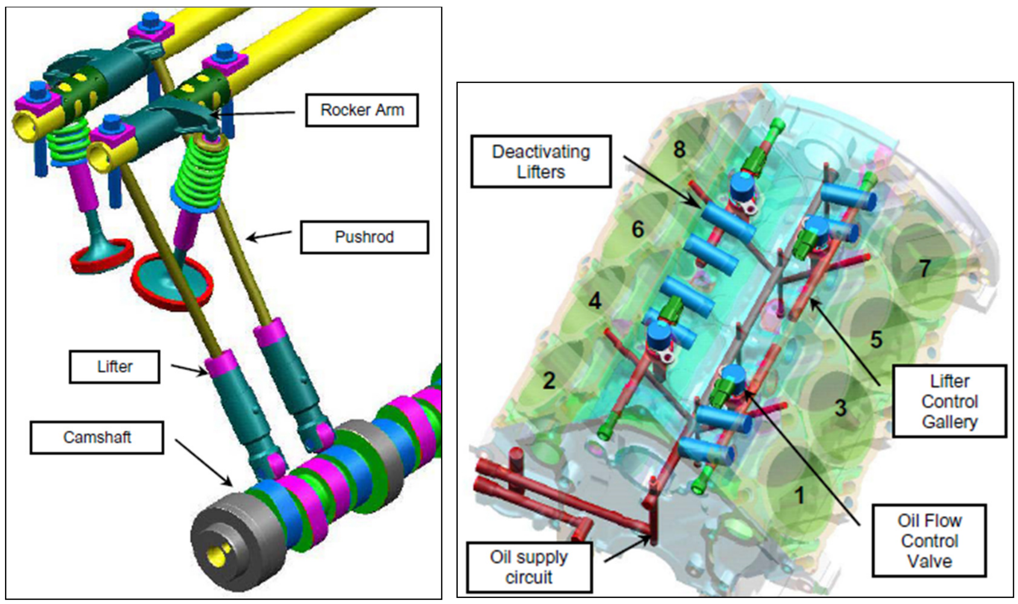


Figure 1.16 : a) The lifter mechanism implemented on pushrod to disconnect cam shaft motion from rocker arm and so from the valve, b) Extra oil channels inside the engine block showing how the lifter mechanism functions [13].

The sequence of valve disconnection is quite important to keep cylinder temperature hot enough for maximizing the efficiency. Therefore, hot exhaust gas should be kept inside the cylinder. Keeping exhaust gas inside the cylinder can be achieved by turning exhaust valve off primarily [13].

1.2.4.2 Skip cycle method

The skip cycle method suggests shutting off the cylinder in some of the consecutive running thermodynamic engine cycles. It can be said that the skip cycle method is a partially cylinder deactivating strategy. Basically, this method is built on the same idea that variable stroke volume method stands on. When the required power is reduced, the effective stroke volume is also reduced by cycle skipping.

The previous section discusses variable stroke volume method in internal combustion engines. To do the same work with reduced number of functional cylinders, non-functional cylinders are totally closed with intake and exhaust valves, functional cylinders shift to a more efficient working region with higher cylinder pressures. Skip cycle method follows the same logic. Normally, a four stroke engine completes a thermodynamic cycle after each 2 revolutions of crank shaft. This means after each 720º crank angle required work is done. However, in skip cycle method this situation changes. Depending on the cycle skipping strategy, after the work gained cycle the consecutive one or two cycles can be skipped without any injection or firing. Therefore in skipped cycles combustion does not happen. Again depending on the number of skipped cycles, work is gained after each 1440º (2×720º) or 2160º (3×720º) of crank angle.

To avoid pumping loss, both the intake and exhaust valves should be totally closed during skipped cycles same as valves are closed for non-functional cylinders in variable stroke volume method. Therefore, the energy is stored inside the cylinder when there is no combustion. This situation can be understood better if the similarity between the squeezed or elongated spring and the totally closed cylinder is discovered. The only loss is due to the friction between the piston and bore interface and among the gas molecules remaining inside the closed cylinder.

If one cycle is skipped after each combustion cycle, then the engine becomes an 8 stroke engine. If consecutive two cycles are skipped after each combustion cycle, then the engine becomes a 12 stroke engine. For each skipped cycle, fuel injection or spark should definitely be cut off and the valves should be closed. Otherwise, engine does work on those skipped cycle cylinders and the local efficiency for those cylinders drops below zero.

Skip cycle method was first proposed by Kutlar (1999) in his Ph.D. thesis in 1999. Then it was patented [14]. He used an inline 4 cylinder spark ignition engine during his experiments. He focused on low load and low speed region of the engine. In his test engine the fuel is injected into the intake manifold and the clean air flow is controlled by the throttle valve. During the experiments some combinations of combustion cycle (N) and skipped cycle (S) were tested. The results with …-NSS-NSS-NSS-… combination at 1200rpm and 1 bar mean effective pressure (low speed and low load) show approximately 10% increase in the efficiency [1].

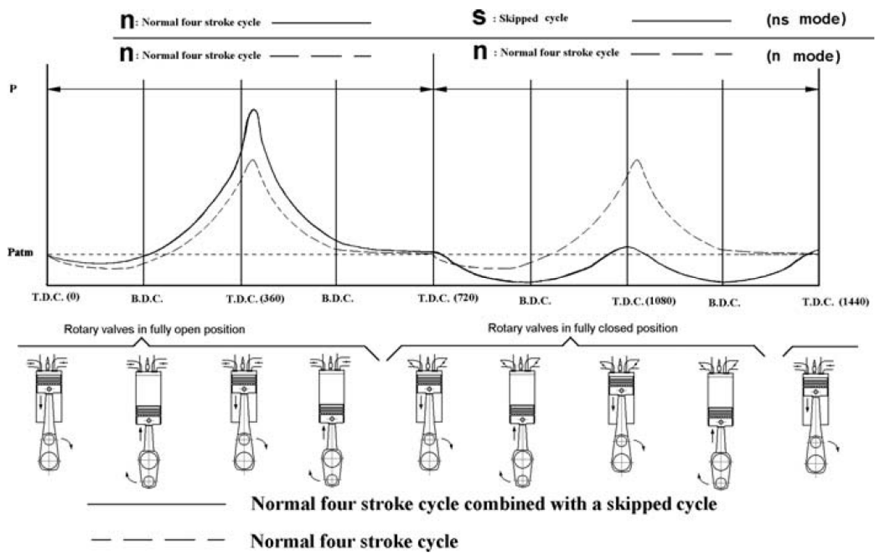


Figure 1.17 : The comparison of conventional working mode of Kutlar's experimental engine with the NS mode on the basis of cylinder pressures. [14]

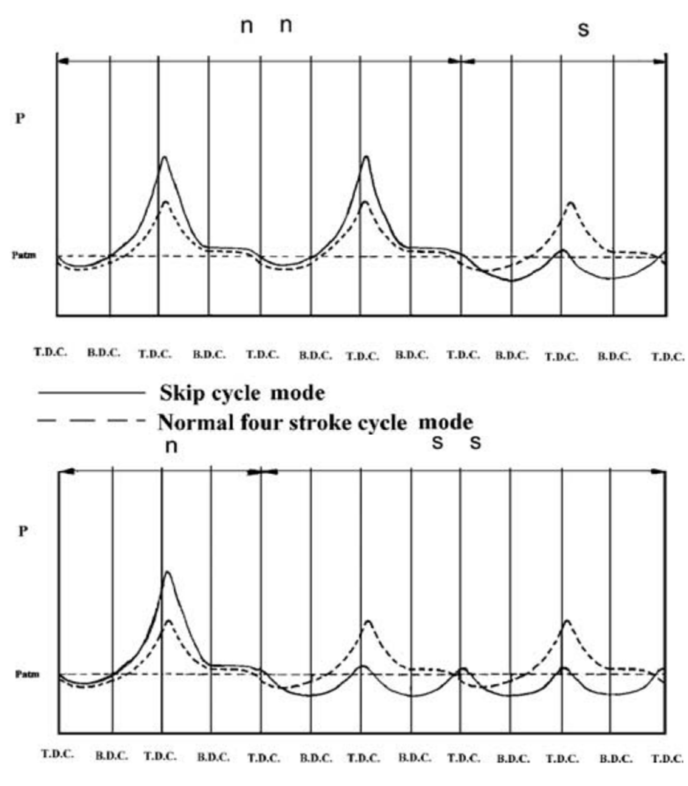


Figure 1.18 : The comparison of conventional working mode of Kutlar's experimental engine with the NNS and NSS modes on the basis of cylinder pressures. [14]

1.3 Literature Review

As expected, there is a continuous development work among automotive technology. This development work has been constituting a great know how and a very big literature.

For example, DaimlerChrysler releases its experiences on designing the most suitable lubrication system for its engines in SAE World Congress in 2007 [15]. In the paper, the quality of an engine lubrication system is clearly defined. The importance of economizing lubrication system is shown. The aim of the work is finding the best strategy to supply just enough oil flow through the engine. In this way, the power consumed by the oil pump is reduced or avoided. DaimlerChrysler's this work shows how much effort is paid to optimize lubrication system.

Knowing the importance of oil consumption levels within the engine and by the engine, many significant studies have been presented for the oil consumption by the engine. Veettil and Shi investigated the gas exchange mechanisms over the piston rings which can also be directive for oil consumption studies [16]. They modeled 88mm piston bore and 98mm piston stroke piston-liner interface. Then CFD analyses were conducted to understand the blow by and oil consumption characteristics. Basically continuity equations, Navier-Stokes equation and the energy equation were occupied for the solutions. The oil consumption characteristics were changed depending on the ring positions in the ring grooves. When the piston rings sit at the bottom surface of their grooves with no instability, their sealing performance gets better compared to instable ring situation. Otherwise, oil is forced to move into the combustion chamber by blow by gas leakage. In Figure 1.19 and in Figure 1.20, the oil consumption values for both piston rings are at the bottom surface of the grooves and are at the middle of their grooves conditions are presented.

Cavallaro et al. emphasizes the importance of the thermal data input for previously mentioned blow by and oil consumption CFD analyses in one of its blow by studies [17]. They explain the blow by path in its technical paper. According to their study, thermal data defines the piston end gap clearance, blow by gas pressures, piston expansions and ring back clearances in the grooves.

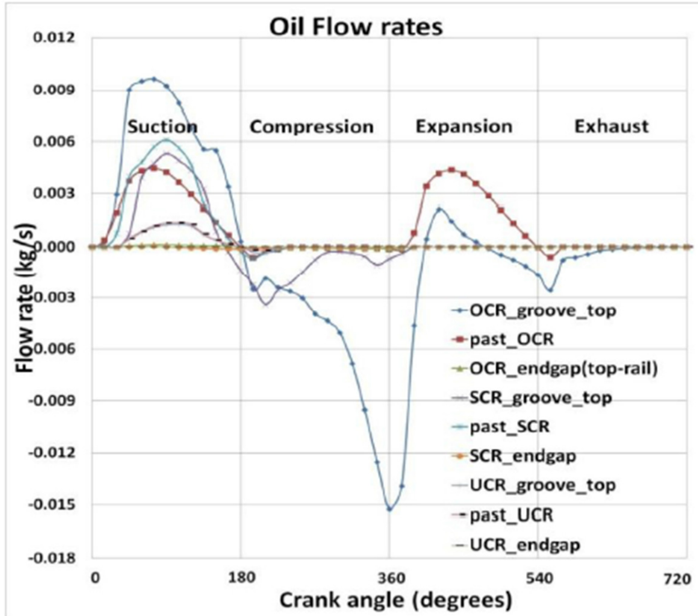


Figure 1.19 : Veettil and Shi CFD analyses result for oil consumption over piston rings. Piston rings sit at the bottom surface of the groove with no instability. [16]

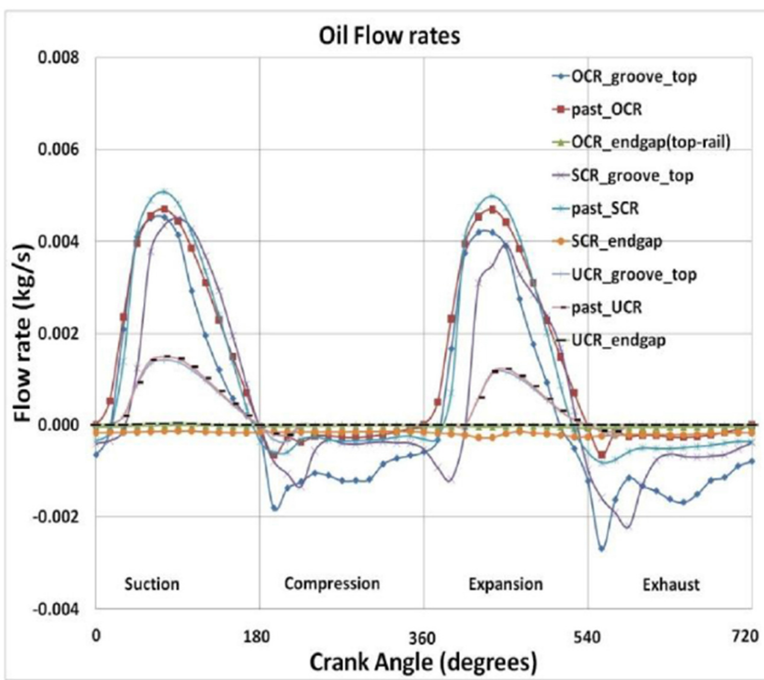


Figure 1.20 : Veettil and Shi CFD analyses result for oil consumption over piston rings. Piston rings are at the middle of the groove. [16]

Satoh, et. al, developed a CFD model to understand the efficiency of oil separator in the engine CCV system [18]. They wanted to predict oil consumption through the CCV system. Their model was used to optimize the oil separator design for the best oil separation efficiency. This technical paper is a good example which shows the effort spent by OEMs to reduce oil consumption by the engine.

Cho and Tian contributed the literature with the studies focusing on oil transportation into the combustion chamber over the piston rings [19]. The outstanding part of this study is that the engine oil was modeled in details of its chemistry. The main assumption with the oil model was that the oil is composed of several different paraffin hydrocarbons with different mass fractions and boiling points. Again their study results show the importance of the surface temperatures which effect oil vaporization. According to the study the main vaporization contributor is the oil around hot top ring area; however, the high gas pressure reduces the vaporization rate. High oil temperature and oil volatility increase vaporization rate where vaporization area reduces due to viscosity reduction. Study validates the vice versa situation as well.

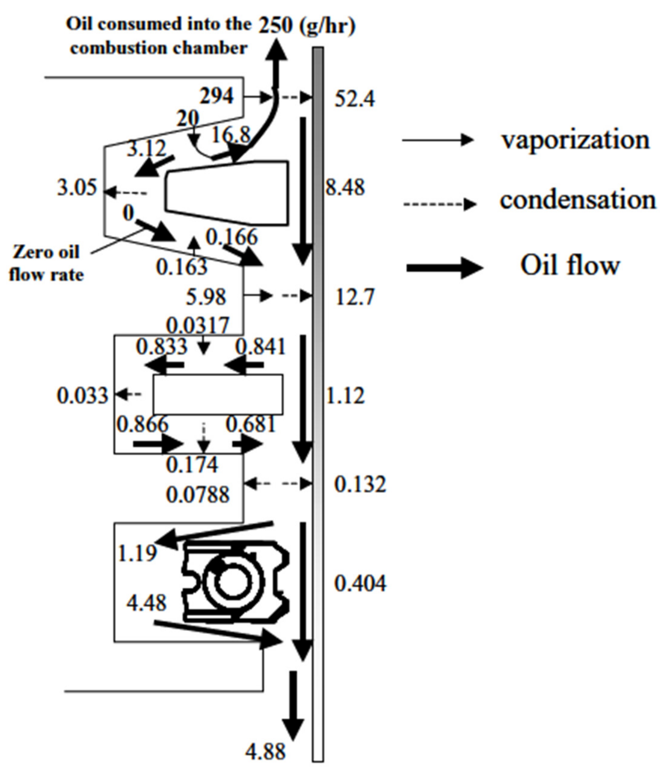


Figure 1.21 : Total oil vaporisation rate and oil vapor transfer rate results of Cho and Tian work with the assumption of fully wetted surfaces. (unit: g/hr) [19]

Xu, et. al, presented the effects of oil consumption on vehicle after treatment system [20]. The main poisoning item inside the oil is ZDDP for the catalyst in the after treatment system. ZDDP includes zinc and phosphorus inside its content and is required to increase wear reduction performance of the lubricant. However, the phosphorus inside ZDDP covers catalyst inner surface and disables the required chemical reactions within the catalyst. Xu, et. al, conducted a series of experiments with different modifications on the engine to understand the impact of oil consumption path on the catalyst deactivation. In Table 1.4 four different steps of the experiment are listed. Those 4 steps were completed with the same highway cycle. High PCV flow which is the second step resulted in the biggest oil consumption. The third step resulted in the medium oil consumption value and the fourth step showed a similar oil consumption amount with the baseline which was the lowest among those four steps. However, the catalyst deterioration levels are not proportional to the oil consumption levels. Step #2 and step #1 had a similar and the lowest deterioration. Step #3 had the highest deterioration level.

Table 1.4 : Xu, et. al, catalyst aging processes through controlled oil consumption routes. [20]

#	Test Name	Oil Consumption Route	Modification
1	Baseline (with no modification)	85-90% from piston rings 10-15% from PCV	1- New high quality valve stem seal 2- Oil through PCV verified by the oil trap at PCV
2	High (open) PCV	50-60% from PCV valve 40-50% from piston rings	1- The inside of the PCV was opened (PCV as a piece of pipe) 2- New high quality valve stem seal 3- Verified by the oil trap at the PCV
3	Open intake valve stem seals	50-60% from intake valve guides 35-45% from piston rings <5% from PCV	1- Cut intake valve seals 2- High quality PCV 3- Verified with an oil trap at the PCV 4- Oil consumption compared with the baseline
4	Open exhaust valve stem seals	Oil consumption similar to the baseline due to the positive pressure in the exhaust line.	1- Cut exhaust valve stem seals 2- Oil trap used at PCV

One of the most recent studies was published in 2012 to explain the lubricant effect on spark ignition engine knock phenomena [21]. Study shows the contribution of the oil present in the combustion chamber to the engine knocking. Amann and Alger suggest usage of low reactivity engine lubricants to reduce knocking for the spark ignition engines.

In 2009 Heywood and Welling published their research results to define the recent trends among automotive SI and CI engines [22]. Their research covers eight years between 2000 and 2008. They say 6% of the automobiles that are powered by gasoline engines have cylinder deactivation technology in North American market. Therefore, it is obvious that the cylinder deactivation technology is a trending topic in automotive sector and can be accompanied by skip cycle cylinder deactivation in the future which is not so far away.

Previously GM cylinder deactivation technology on 5.7L HEMI® was mentioned. GM changes total engine displacement volume by shutting down pre-specified cylinder valves [13]. The pre-specified cylinders are deactivated and no combustion occurs inside those cylinders when they are deactivated. According to Falkowski, et. al, cylinder deactivation improves fuel economy with the challenges of cost and NVH.

Many other engineers worked on cylinder deactivation idea as Falkowski, et. al, did. Fujiwara, et. al, is an example of those engineers. However, they bring an innovation to this system. Usually, half of the cylinders are deactivated at the same working period. Their engine can deactivate 3 of 6, 4 of 6 or all 6 cylinders [23]. They claim that their new variable cylinder management technology achieves 10% fuel economy on their new 3L V6 gasoline engine compared to the old version 3L SOHC VTEC® engine.

Leone and Pozar released cylinder deactivation investigation results in 2001. The paper shows the fuel economy benefit of cylinder deactivation strategy in various operating modes such as idle, low engine speed, 1st and 2nd gear and warm up after cold start [24]. Also the effects of engine size and vehicle weight on the fuel economy benefit were presented by Leone and Pozar. The results were produced by a CAE simulation tool. They claim up to approximately 10% fuel economy by cylinder deactivation. In general, the benefit is bigger in city driving cycle. Also it varies depending on the driving cycle for US, Europe and Japan. On the basis of vehicle attributes, cylinder deactivation offers the biggest fuel economy benefit for high performance vehicles. In Table 1.5 the summary of the results that were generated by Leone and Pozar to understand the cylinder deactivation benefit are listed.

Table 1.5 : Variable displacement engine fuel economy benefit (%) for various vehicles, engine displacements and test cycles [24].

	Vehicle	Displacement [L]	EPA City	EPA Highway	EPA M/H	NEDC	Jap. 10-15
2722kg light truck with 5	A	5,40	8,80	5,20	7,50	7,60	8,90
speed auto transmission	A	6,80	11,70	9,90	11,00	10,20	11,40
1928kg pascar with 4	B	4,60	9,90	7,80	9,10	9,50	7,30
speed auto transmission	B	5,40	11,60	10,70	11,30	10,60	9,10
2381kg light truck with 4	C	5,40	8,00	2,90	5,90	7,00	7,10
speed auto transmission	C	6,80	11,50	7,80	10,00	9,80	10,10

Luxury car industry is also affected by this cylinder deactivation trend. Senapati et. al. released their cylinder deactivation studies in 2011 with a V8 engine [25]. They collected CO_2 reduction benefits of cylinder deactivation technology. They also claim that fuel economy improvement is achieved without sacrificing any attribute for vehicle refinement performance.

Fiorenza, et. al. searched the benefit of cylinder deactivation technology in smaller passenger car engines in 2003 [26]. They combined the cylinder deactivation idea with variable valve timing technology to increase the gain of cylinder deactivation application. The experiments were conducted on a vehicle which has a port deactivation system and a continuously variable cam phaser system on the manifold injected gasoline engine. They claim that up to 6% fuel economy can be achievable in the NEDC test cycle together with cylinder deactivation and variable cam timing system.

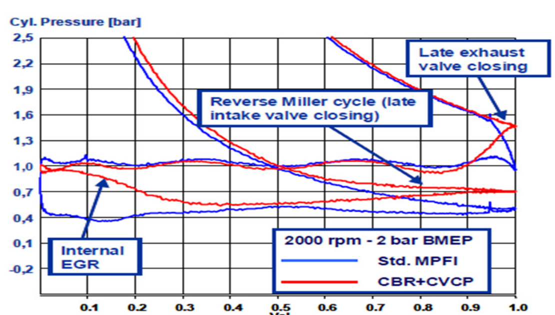


Figure 1.22 : Reduction of losses, that are sourced by the pressure drop across the throttle, at part load [26].

Table 1.6 : Vehicle test results of Fiorenza, et. al, work based on different engine configurations [26].

	MPFI 1.2L 16V (Baseline)	PDA+CVCP 1.4L 16V	MPFI 1.4L 16V
Test: ECE/EUDC cold			
Fuel Consumption (L/100km)	6.02	5.77	6.17
HC emission (g/km)	0.072	0.074	0.075
CO emission (g/km)	0.27	0.43	0.65
Nox emission (g/km)	0.042	0.057	0.045
Test: ECE15 cold			
Fuel Consumption (L/100km)	7.95	7.44	8.11

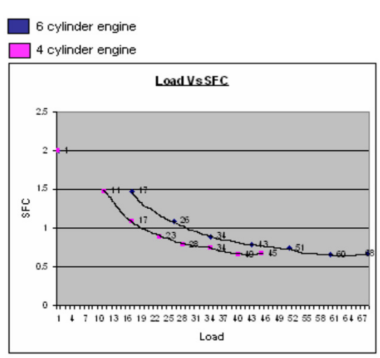
In 2005 two different studies were published. Both were discussing an alternative method which was called controlled auto ignition for fuel economy. Controlled auto ignition combines the advantages of both spark ignition and compression ignition engines. Premixed air and fuel mixture can be sent into the cylinders and then combustion occurs due to compression. The goal of this application is using the residual energy after the combustion. Therefore, fuel economy and high level of NOx emission reduction are expected. One of the studies was completed by Dougles et. al. in 2005 [27]. In their experiments, both controlled auto ignition and cylinder deactivation strategies were applied. They say 75% of NEDC allows application of controlled auto ignition and cylinder deactivation [24]. Dougles et. al. claims 10% fuel economy and 28% NOx emission reduction on NEDC with their study. The other study was published by Lang, et. al [28]. In this study both the fuel economy and the emission reduction benefits expectances are much bigger. They say 15% fuel economy is achievable with controlled auto ignition on hot NEDC [28].

Kutlar et. al. brings a different view on cylinder deactivation strategy [14]. He proposes skipping consecutive thermodynamic cycles and changing engine working mode of 4 strokes to 8 or 12 strokes. Therefore, general perception of gaining work after 4 engine strokes is destroyed by this new approach. Consequently, the required work can be gained after each 8 or 12 strokes depending on the cycle skipping strategy. His experiments resulted in up to 50% indicated efficiency increase at low load conditions. It should be noted that, Kutlar et al worked on a small displaced volume passenger car gasoline engine.

However, the working pattern for internal combustion engines proposed by Kutlar has already been discovered by Henein et al. After a series of tests conducted in cold ambient conditions two different but related papers were published by Henein et al. One of the papers was focusing on combustion instability during diesel engine cold starting [29]. In this paper it is claimed that misfiring, in other words cycle skipping, is a repeatable phenomenon during diesel engine cold starting rather than a random action depending on the compression ratio, fuel volatility and the temperature. The engine may start with 12 strokes or 8 strokes and then by time turns into regular 4 stroke operation mode. As the temperature increases, skipped cycle quantity reduces. In the following paper in 1990, they combed through the issue and analyzed actual thermodynamic cycles of an air cooled single cylinder 4 stroke experimental diesel engine at cold ambient conditions [30].

In 2005, Vinodh tried cylinder deactivation on a diesel engine [31]. His idea was changing the total displaced volume by decoupling crank shaft portion which was connected to the deactivated cylinder pistons. Therefore, power loss due to piston friction is avoided at the deactivated cylinders by this method. However, it should be noted that the hardware which enables the crank shaft to be partially controlled is a complex system so that needs time to be commercialized. Nevertheless, Vinodh's proposal is a beneficial model to increase diesel engine efficiency at part loads by deactivating some cylinders.

a)



b)

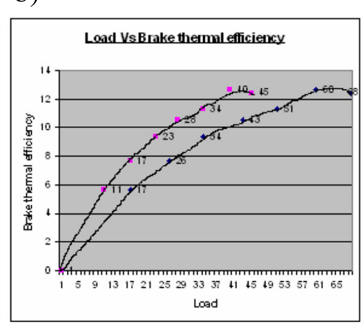


Figure 1.23 : a) SFC reduction, b) Efficiency increase [31]. Pink dots are 4 cylinder working mode and the other curve is 6 cylinder working mode

Finally, Ma came with a study on oil consumption over piston rings, when cylinder is deactivated, in 2010 [32]. Ma's work shows when the cylinder is deactivated the vacuum inside the combustion chamber creates a pressure gradient which forces oil to be transported into the cylinder towards the combustion chamber, as the ring end gaps are bigger due to reduced temperatures oil transportation into the combustion chamber becomes easier, also fluttering top ring pushes the oil existing in its groove to the top land of the piston; however, scraped by the top ring amount is reduced. Nevertheless, the total amount of oil transferred into the combustion chamber is bigger in deactivated cylinder than it is in active cylinder. Ma proposes reducing the clearances between piston rings and the groves, ring end gap clearances, cylinder bore distortions, revising piston top land design and limiting the overall oil supply to the deactivated cylinder pistons to reduce transported into the combustion chamber oil amount for deactivated cylinders [32].

1.4 Objective

All information that was given formerly is presented to explain the technological development history of efficiency increase for internal combustion engines and the reason which pushes engineers to work on efficiency increase developments. The importance of minimizing any kind of consumption within the engine was explicitly discussed. Engine lubricant is one of those consumables.

Oil consumption can be divided into two categories. Oil consumed by lubricated components such as bearings, pistons or valve-train components. This kind of oil consumption means the oil flow requirements through the engine. The other meaning is the reduction in total oil amount inside the engine. This thesis focuses on oil consumption inside the combustion chamber which results in burning the oil and scavenging out of the engine by exhaust.

The aim of this study is basically to understand oil transportation characteristics over piston rings when the cylinder actively functions and is deactivated. Cylinder deactivation is applied with vacuum pressure while simulating thermodynamic cycle on a natural aspirated gasoline engine model. During the study, it is assumed that the cylinder deactivation occurs just after the exhaust stroke of a successfully completed active cylinder thermodynamic cycle. Therefore, all valves are assumed to be shut down just before the consecutive intake stroke for deactivation.

Although there are various oil consumption investigations in the literature, as the cylinder deactivation is a recent technological development, very limited information about deactivated cylinder oil consumption exists. This study investigates the oil consumption behavior of a single cylinder diesel engine in different cycles such as normal running condition, fully closed and passivized condition and skip cycle mode. Cycle skipping results in vacuum pressure inside the cylinder as explained in the previous paragraph. The effects of this vacuum on oil consumption are investigated.

2. THEORY AND MODELING

2.1 Production Of Thermodynamic Data For Piston And Liner Model

The cylinder pressures and gas temperatures are quite important parameters for oil consumption calculations through piston and liner interface. Basically they are indicative for oil evaporation over the surface of oil film covering the liner and thrown off by top ring oil amount into the combustion chamber. Therefore, the more precise thermodynamic input is the more precise oil amount transported into the combustion chamber result.

The thermodynamic data was produced in the software RICARDO Wave. This program is useful to get realistic engine performance data from one dimensional model. 1D model results are based on conservation of mass, conservation of energy and conservation of momentum rules. The time step applied during calculations should obey Courant condition. Otherwise, the solutions cannot converge [1].

The mechanism of the skip cycle cylinder passivation method has already been discussed in the introduction chapter. 8 stroke working condition can be defined for the model by using built in program functions in RICARDO Wave. However, the idea of cylinder deactivation is basically includes manipulating valve motions as well to avoid pumping loss for the cylinder deactivated engine when the valves are fully closed. Therefore, a special valve control algorithm is adapted into the engine model [1]. The two differences between 4 stroke conventional running condition and the 8 stroke skip cycle condition are the stroke quantities and the diameter of the Orifice_1 at the Figure 2.1.

The data has been produced with the following assumptions [1]:

- As the comparison is done between working cylinder and deactivated cylinder, there is no need for a real engine correlation.
- The same combustion parameters were occupied in different models.
- Part load for a conventional 4 stroke gasoline engine is simulated by an orifice located in the intake side which can be decreased in diameter.

- The excessive air multiplier is taken as 1 for part load condition.
- Torque target is kept as same for skip cycle calculations to obtain a fair comparison between break specific fuel consumption values.
- Frictional constants is kept same from normal running 4 stroke cycle to skip cycle 8 stroke running condition.
- The combustible mixture which is prepared in the intake manifold is assumed to be homogenous.

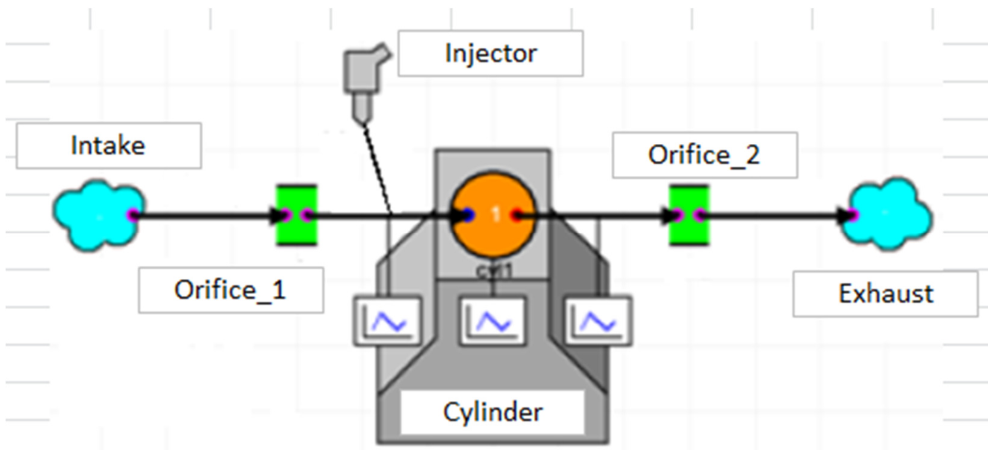


Figure 2.1 : Engine model that was used to generate cylinder pressure and gas temperature data for oil transport calculations [1].

The valve control algorithm embedded into the model enables manipulating valve and injection timing. Then it is possible to control the combustion for skip cycle simulation. To run a stabilized skip cycle simulation, firstly model functions as a conventional 4 stroke engine until it is stabilized and then cycle skipping operation starts. At the beginning the cycle quantity is given as an input to the system to let the model differentiate the combustion cycle from the skipped cycle. According to the results of differentiation process which is done by skip cycle formula in Figure 2.2, the injection, valve and combustion switches function.

As a result, when the cycle is skipped, the output of the skip cycle formula becomes zero and so are the valve, injection and combustion control elements are. Then the valve switches close the valves, the injection switch cuts off the injected fuel. Combustion switch can be perceived as an extra control element for a safer simulation. It shifts the operation from conventional run to skip cycle. When the model simulates conventional 4 stroke engine the combustion switch is 1 and when it simulates skip cycle condition the combustion switch is zero. This means combustion switch disables or enables spark plug according to the operation.

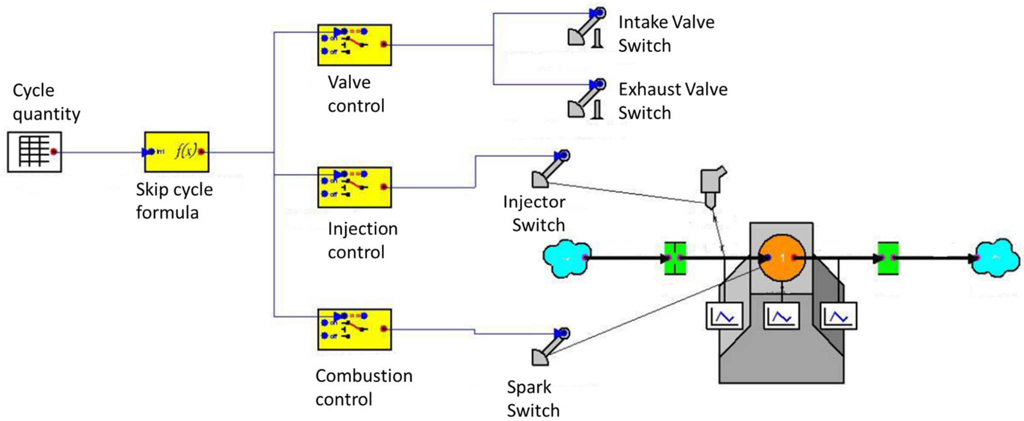


Figure 2.2 : Skip cycle control algorithm is embedded into the single cylinder engine model [1].

Finally, using the one dimensional single cylinder gasoline engine model cylinder pressure data as a function of crank shaft angle is produced by application of some mathematical equations. They are [1]:

$$s(\alpha) = (r + l) \cdot \cos\Psi - r \cdot \cos(\Psi + \alpha) - l \cdot \sqrt{1 - (\frac{r}{l} \cdot \sin(\Psi + \alpha) - \frac{e}{l})^2} \quad \textbf{(2.1)}$$

$$\Psi = \sin^{-1}(\frac{e}{r + 1}) \quad \textbf{(2.2)}$$

$$V(\alpha) = \frac{\pi \cdot D^2}{4} \cdot s(\alpha) \quad \textbf{(2.3)}$$

$$V_s = \frac{\pi \cdot D^2}{4} \cdot s \quad \textbf{(2.4)}$$

$$V_b = \frac{V_b}{\varepsilon - 1} \tag{2.5}$$

$$V_{volume\ ratio}(\alpha) = \frac{V(\alpha) + V_b}{V_b} \tag{2.6}$$

$$IMEP = \frac{\sum_{Cycle\ start}^{Cycle\ end} (P_n + P_{n+1}) \cdot (V_{n+1} - V_n)}{2 \cdot V_s} \tag{2.7}$$

$$GMEP = \frac{\sum_{-180}^{+180} (P_n + P_{n+1}) \cdot (V_{n+1} - V_n)}{2 \cdot V_s} \tag{2.8}$$

$$PMEP = IMEP - GMEP \tag{2.9}$$

where ;

s is the piston stroke (mm),

r is the crank shaft radius (mm),

l is the connecting rod length (mm),

e is the piston pin offset (mm),

α is the crank angle (º),

Ψ is the angle between crank shaft vertical position and TDC position (º),

V_s is cylinder volume (cm^3),

V_b cylinder space volume (cm^3),

ε is the compression ratio,

$V_{volume\ ratio}$ is the volume ratio,

IMEP is the indicated mean effective pressure (Pa), GMEP is the gross mean effective pressure (Pa),

PMEP is the pumping mean effective pressure (Pa),

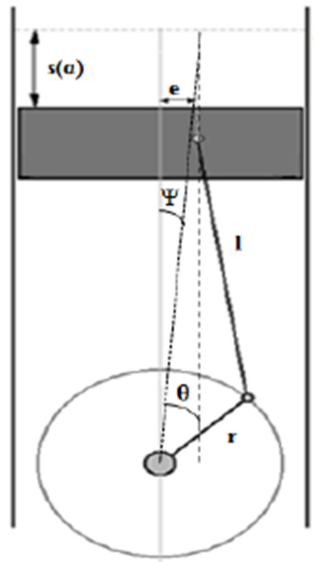


Figure 2.3 : Generic engine power conversion system overview for a better understanding of the given formulation [1].

It should be noted that, for different cycle skipping strategies cycle end point changes. Due to the shift at the cycle end point, all mean effective pressure calculations are done through whole cycle. For example, in conventional engine simulations GMEP is calculated from -180º crank angle to +180 º crank angle ; however, if one cycle is skipped (…NS…) the positive or negative work gained during skipped cycle compression and expansion strokes should be added on top of that GMEP [1].

In following sections it will be seen that the skipped cycle thermodynamic data is used to calculate the oil consumption through piston assembly in the case of fully deactivated and shut down cylinder. Also, the firing first 4 strokes of that produced thermodynamic data is used to simulate active cylinder oil consumption.

2.2 Theory Of Oil Transport Through Piston And Liner Interface

This section is almost totally adopted from AVL Excite Piston and Rings Module Theory [9]. While studying oil consumption in different working cycles of the regarding one cylinder engine model, AVL Excite program Piston Ring Dynamics and Lube Oil Consumption Module was used.

Blow by gas flow rate, lube oil consumption and friction losses of the piston liner group are strongly dependent on piston ring movements. This is why piston ring dynamics calculations are so important. Ring dynamics are influenced by both the piston radial and tilting motions. Oil consumption mechanisms through piston, ring and liner system are as follows:

- Evaporation at the oil film from the liner surface.
- Oil transportation around the first piston ring, including oil throw off.
- Oil blow through the gap of the first ring from and into the combustion chamber.
- Oil scraping at the top land's top edge considering deposits.

The oil film thickness variations on the liner surface depending on thrust and anti thrust sides should also be taken into account while calculating the oil consumption.

The oil coating cylinder wall is exposed to high temperatures and high gas flow rates. Vaporized oil due to the stationary convective material exchange between lube oil and combustion gas becomes a part of cylinder gas and is assumed to leave the engine as either partially burned or unburned hydrocarbons. The evaporation rate of the lube oil from the liner surface can be calculated with the help of the below equation [9]:

$$\dot{m}_{evap.\ Oil} = \beta \cdot \left(c_{film} - c_{\infty}\right) = -D_c \cdot \frac{dc}{dx} \qquad \textbf{(2.10)}$$

where

$\dot{m}_{evap.\ Oil}$ is mass flux through boundary surface [kg/m^{2}h]

β is material transmission coefficient [m/h]

c_{film} is concentration of lube oil at film surface [kg/m^3]

c_{∞} is concentration of lube oil in combustion chamber [kg/m^3]

D_c is diffusion coefficient [m^2/h] x is the coordinate perpendicular to boundary surface

Δs is the movement of the piston during time step Δt

s_{film} is uncovered area of the oil film

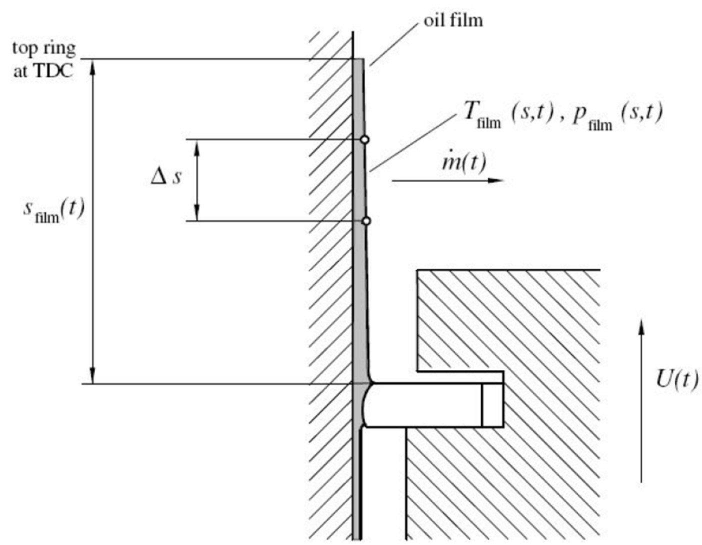


Figure 2.4 : Oil mass flux into combustion gas [9]. Oil film thickness change can be seen around the top ring in the figure.

Figure 2.5 can be refered for the calculation of oil film surface temperature (T_{film}) with the assumption of pure oil vapor layer existance over the oil film surface.

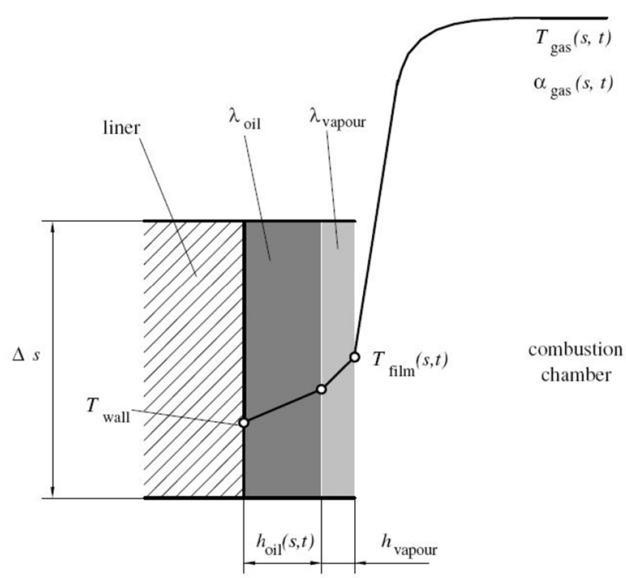


Figure 2.5 : Oil evaporation process schematic [9]. The curve shows the temperature gradient from gas to the wall.

Recalling ideal gas equation, mass fraction of the species inside the combustion gas can be calculated from the Equation 2.11 [9].

$$c = \frac{p}{RT} \tag{2.11}$$

where

p is the partial pressure [N/m^2]

T is gas temperature [K]

R is gas constant [j/kgK]

Substituting Equation 2.11 into Equation 2.10 yields to Equation 2.12 [9].

$$\dot{m}_{evap.\,Oil} = \frac{\beta}{RT} \cdot \left(p_{film} - p_{\infty} \right) = -\frac{D_c}{RT} \cdot \frac{dp}{dx} \tag{2.12}$$

where

p_{film} is the oil vapor pressure over the oil film at the liner surface [N/m^2]

p_{∞} is the oil vapor pressure inside the combustion gas [N/m^2]

As the oil vapor amount is negligible inside the combustion gas, oil vapor partial pressure (p_{∞}) can be assumed as zero. And then integrating oil mass flux by the film surface area results in total evaporated mass in one combustion cycle.

One other oil consumption mechanism through piston and cylinder interface is the oil throw off by the piston top ring. Oil throw off is the mechanism where the liquid oil on the cylinder bore surface is transported mechanically into the combustion chamber by acceleration and decelerarion of piston and top ring assembly [7]. The thrown off oil amount can be calculated considering the flow balance of lube oil around top ring and the piston acceleration.

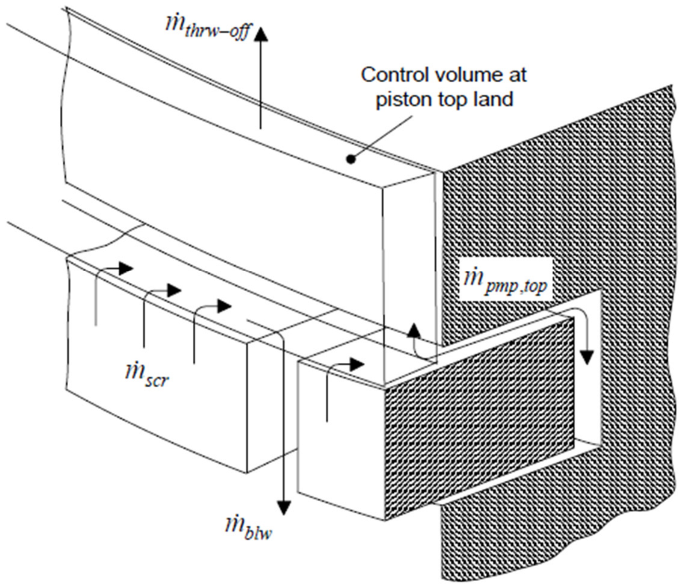


Figure 2.6 : Oil throw off mechanism at piston top land [9]. Control volume at piston top land is the passage for the thrown off oil.

In Figure 2.6 :

$\dot{m}_{thrw\text{-}off}$ is oil mass flux thrown off onto piston surface [kg/h].

$\dot{m}_{scr}$ is oil mass flux scraped by the top ring [kg/h].

$\dot{m}_{pmp,top}$ is oil mass flux over the top side of the first compression ring [kg/h].

$\dot{m}_{blw}$ is oil mass flux through the top ring closed gap into the first inter ring [kg/h].

In this mechanism the total thrown off oil amount is calculated by a simple addition and substraction [9]:

$$m_{acc}(t) = (\dot{m}_{scr} + \dot{m}_{pmp,top} - \dot{m}_{blw})\Delta t \quad \textbf{(2.13)}$$

where $\dot{m}_{acc}$ is total transfered oil into the combustion chamber in the time interval Δt.

Scraped oil amount is the difference between the oil quantities on bore surface for conditions of piston travelling up to TDC and down to BDC. Please see the below equation to calculate scraped oil amount [9]:

$$\dot{m}_{scr} = \text{Max}\left[\left[(h_s + h_{s+\Delta s})_{upward} - (h_s + h_{s+\Delta s})_{down}\right]\frac{\rho\ D\ \pi\ \Delta s}{2\ \Delta t}, 0\right] \quad \textbf{(2.14)}$$

with h_s left oil film of the top ring on the cylinder surface, ρ the density of engine oil, D the cylinder bore diameter, Δs stroke increment and Δt time increment.

Most of the time the combustion chamber pressure is higher than the first inter ring volume pressure. This is basically why a continuous blow by gas flow occurs. This blow by gas flow also transfers some oil on bore surface from piston top land to first inter ring volume. And top ring back pressure which also supports first ring for better sealing function pulses the oil accumulated on top ring upper surface into the combustion chamber when the combustion chamber pressure gets lower.

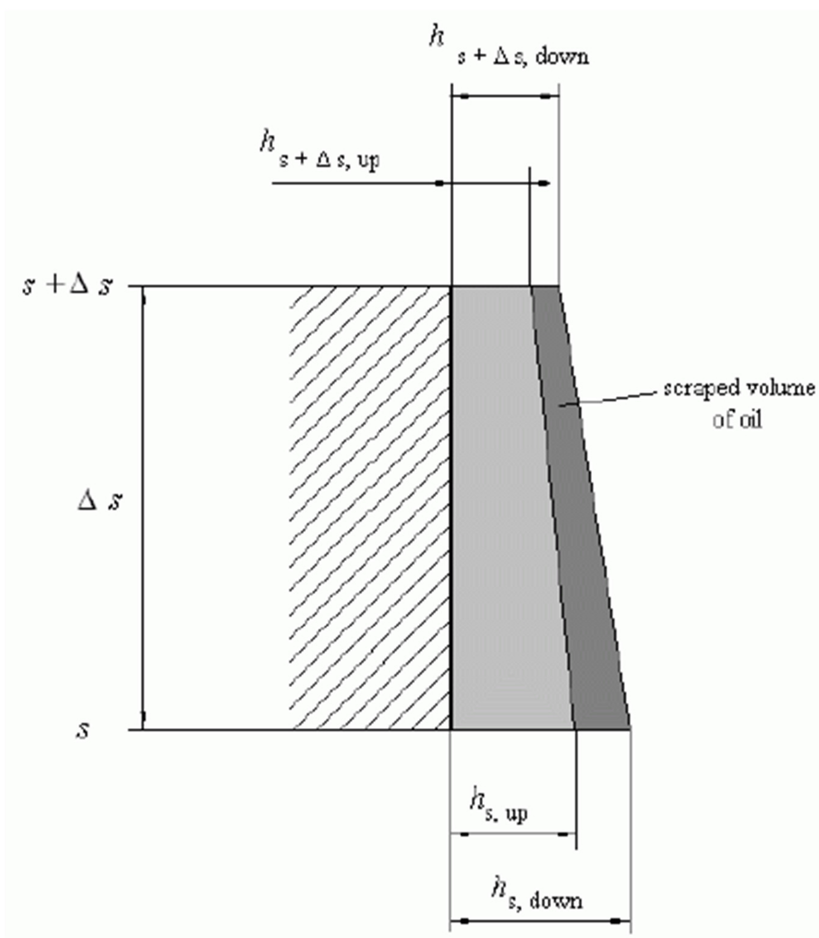


Figure 2.7 : Oil film thickness difference on bore surface between up and down stream movement of the piston [9].

The other oil consumption mechanism between piston and liner interface is the oil blow through top ring close (end) gap. When negative pressure gradient occurs over the top ring from first inter ring volume to combustion chamber, oil is blown through the top ring close gap. This is an instantaneous oil loss and is regardless with the inertial forces.

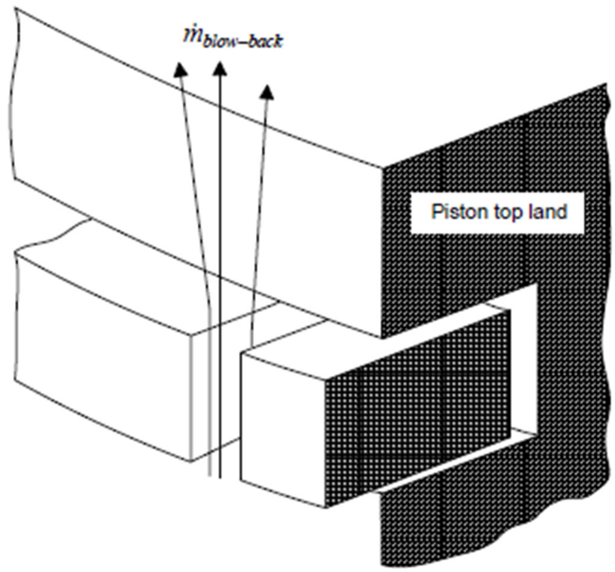


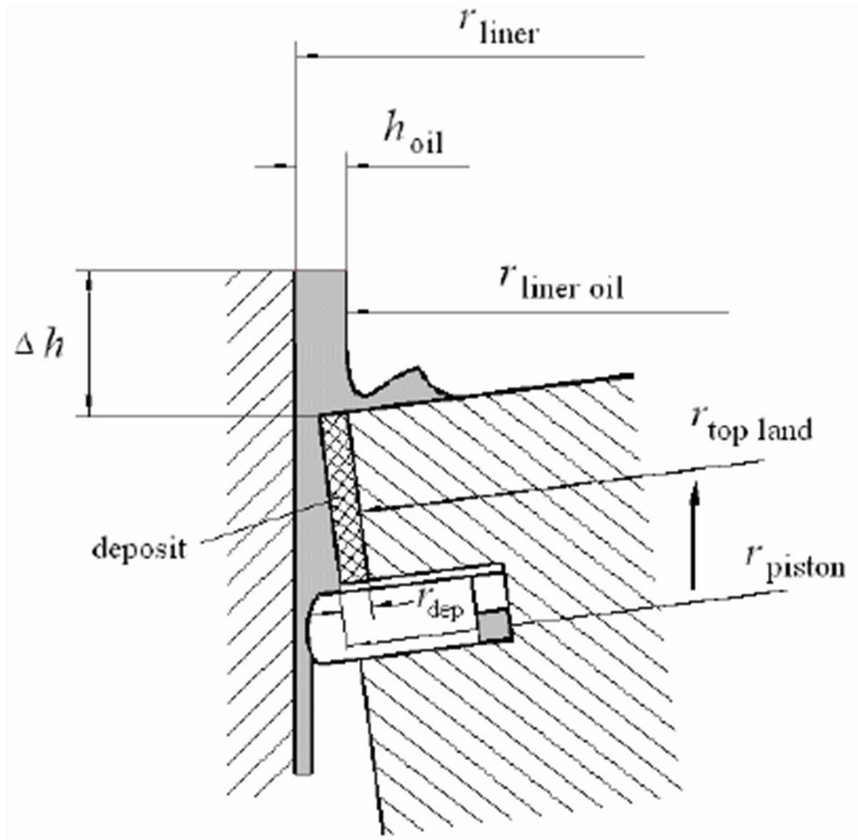
Figure 2.8 : Reverse blow by effect on oil transportation into the combustion chamber [9].

$$\dot{m}_{blow-back} = f_{blow-back} \frac{a^2 \rho}{8 \pi \eta b} Max[(p_{1/2} - p_c), 0] \qquad \textbf{(2.15)}$$

Equation 2.15 is used to calculate oil mass flux into the combustion chamber by reverse blow by gas flow ($\dot{m}_{blow-back}$). $f_{blow-back}$ is the proportional constant which is an ampiric value produced by AVL. α is the actual area at the ring end and b is the width of ring running face. p_c is the cylinder pressure and $p_{½}$ is the inter-ring pressure.

The last oil consumption mechanism between piston and liner interface is oil scrapping of the piston top land edge into the combustion chamber. This scenario is an unusual case for piston movement. Power conversion system is designed to minimize this tilting movement to protect liner against scratches done by piston top land edge.

a)



b)

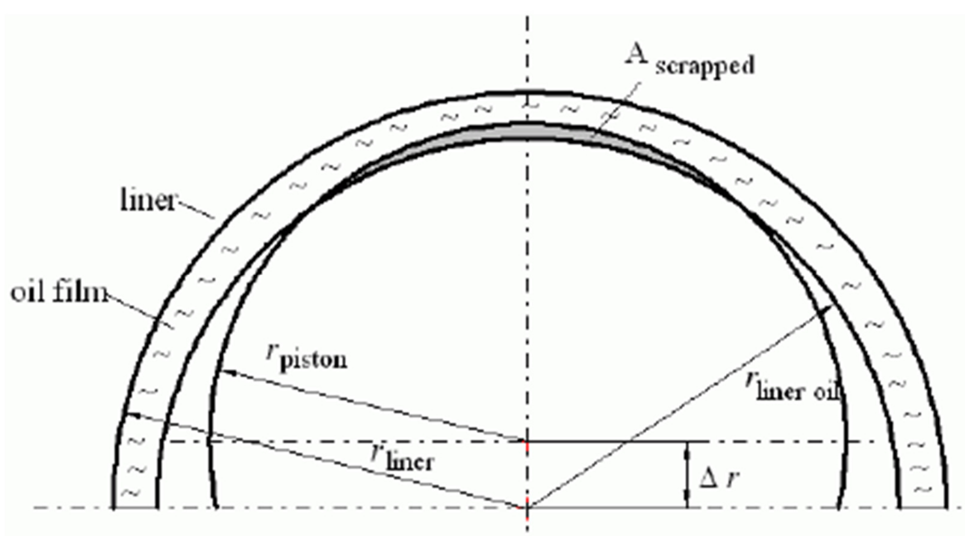


Figure 2.9 : a) Oil scrapping of the piston top land edge into the combustion chamber. b) Piston top land edge and oil film on bore surface overlapping. [9]

In Figure 2.9 ;

r_{liner} is the liner radius [m]

h_{oil} is the oil film thickness [m]

$r_{liner\ oil}$ is the radius of oil coated liner surface [m]

r_{piston} is the piston radius [m]

$r_{top\ land}$ is the piston top land radius [m]

$r_{deposit}$ is the combustion deposits coated top land surface radius [m]

$A_{scrapped}$ is the area of piston top land edge and oil film thickness intersection [m2]

and the scrapped oil volume is a simple multiplication of Ascrapped and stroke increment.

However, considering piston lateral movements with minor tilting motions oil scrapping of the piston top land edge is one of the oil loss mechanisms through piston and liner interface.

2.3 Oil Transport Modeling Of Single Cylinder Piston And Liner Interface

The thermodynamic data was initially produced for 1.6L inline 4 cylinder naturally aspirated spark ignition engine. Therefore, the oil consumption model was adapted to simulate a 1.6L and 4 cylinder engine. 86mm diameter bore and 69mm stroke values were occupied in the oil consumption model basically.

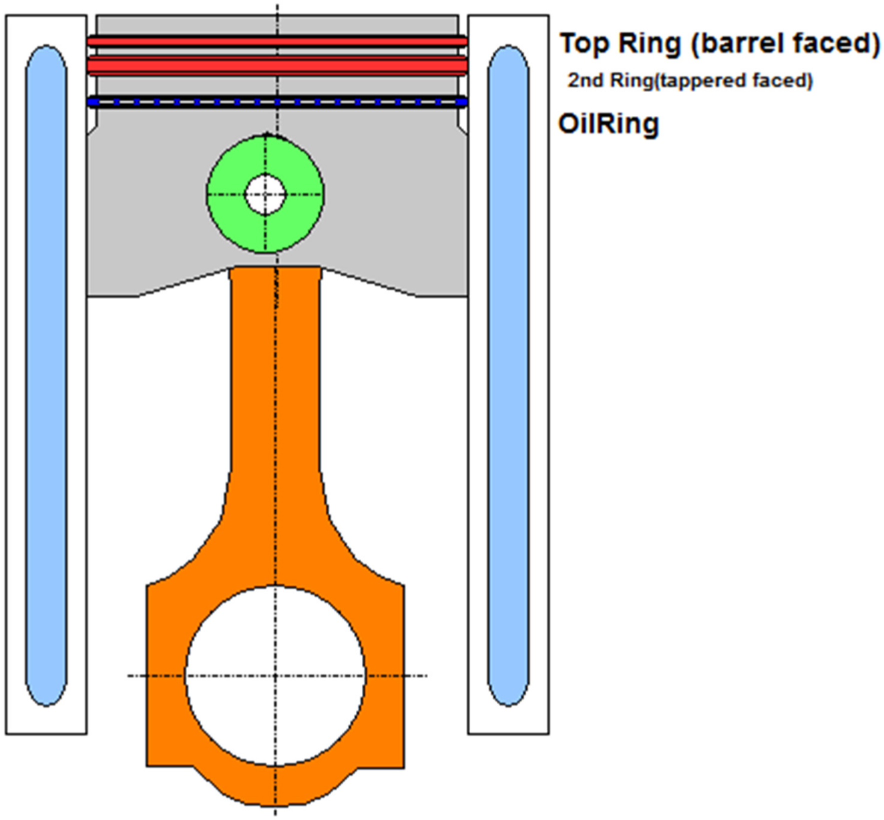


Figure 2.10 : AVL Excite Piston and Ring Dynamics model for oil consumption calculations.

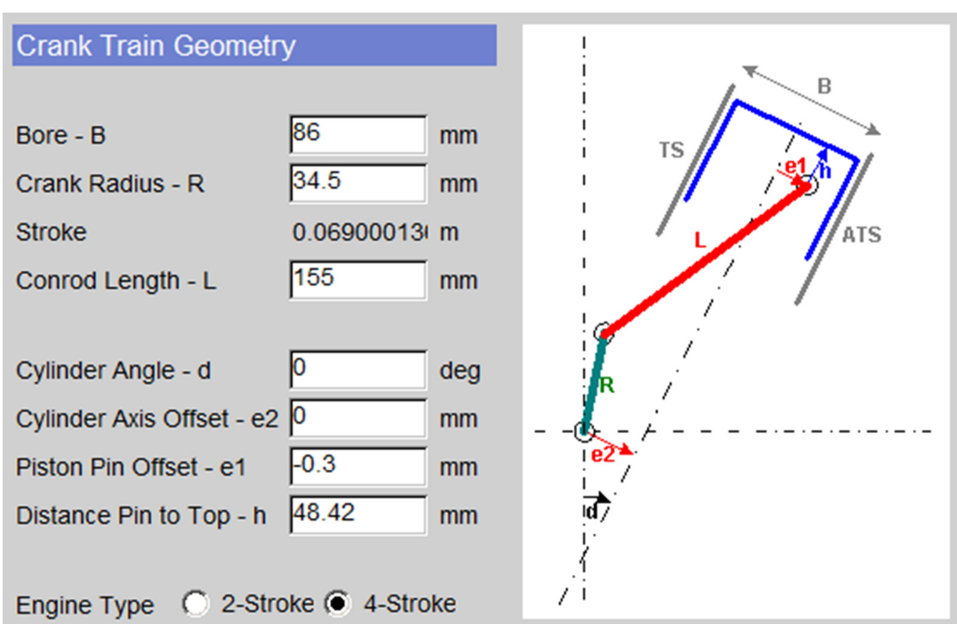


Figure 2.11 : Crank-train details of the oil consumption calculation model in AVL Excite Piston and Ring Dynamics Module.

In Figure 2.10 crank train geometry details are presented for the single cylinder model in order to calculate oil consumption characteristic changes among working conditions. And, in Table 2.1 the details of oil consumption model parameters are presented.

Table 2.1 : Single cylinder oil consumption through piston asy and liner interface model input parameters for AVL Excite.

Parameter	Type	Value	Unit
Closed_Gap_1st_Ring	global	0.05	mm (Length)
Closed_Gap_2nd_Ring	global	0.05	mm (Length)
Closed_Gap_Top_Ring	global	0.05	mm (Length)
DATA_PATH	global		
Groove_Root_Depth_1st_Ring	global	-3.55	mm (Length)
Groove_Root_Depth_2nd_Ring	global	-4.35	mm (Length)
Groove_Root_Depth_Top_Ring	global	-4.35	mm (Length)
Groove_Temp_1st_Ring	global	180	degC (Temperature)
Groove_Temp_2nd_Temp	global	220	degC (Temperature)
Groove_Temp_Top_Ring	global	240	degC (Temperature)
Liner_Surf_Temp_25	global	140	degC (Temperature)
Liner_Surf_Temp_50	global	160	degC (Temperature)
Liner_Surf_Temp_90	global	250	degC (Temperature)
Liner_Surf_Temp_Bottom	global	120	degC (Temperature)
Liner_Surf_Temp_Top	global	280	degC (Temperature)
Liner_Temp_25	global	130	degC (Temperature)
Liner_Temp_50	global	150	degC (Temperature)
Liner_Temp_90	global	240	degC (Temperature)
Liner_Temp_Bottom	global	110	degC (Temperature)
Liner_Temp_Top	global	270	degC (Temperature)
Liner_exp_coeff	global	1.15e-005	1/K (Heat Expansion Coefficient)
Nom_Rad_Cle	global	10	micron (Length)
Piston_Exp_Coeff	global	2.3e-005	1/K (Heat Expansion Coefficient)
Piston_Temp_1Land	global	265	degC (Temperature)
Piston_Temp_2Land	global	240	degC (Temperature)
Piston_Temp_3Land	global	225	degC (Temperature)
Piston_Temp_Bottom	global	130	degC (Temperature)
Piston_Temp_Skirt_D2	global	150	degC (Temperature)
Piston_Temp_Skirt_DN	global	140	degC (Temperature)
Piston_Temp_Skirt_up	global	160	degC (Temperature)
Piston_Temp_Top	global	300	degC (Temperature)
Rad_Dev_1st_Ring_Above_Groove	global	-0.608	mm (Length)
Rad_Dev_2nd_Ring_Above_Groove	global	-0.428	mm (Length)
Rad_Dev_Top_Ring_Above_Groove	global	-0.8535	mm (Length)
Side_Clear_2nd_Ring	global	0.14	mm (Length)
Side_Clear_Top_Ring	global	0.14	mm (Length)
Side_clear_Oil_ring	global	0.07	mm (Length)
SummitR	global	0.03	micron (Length)
Tangential_Force_1st_Ring	global	21	N (Force)
Tangential_Force_2nd_Ring	global	18.35	N (Force)
Tangential_Force_Top_Ring	global	14.9	N (Force)
speed	global	2000	rpm (Angular Velocity)

Table 2.2 : Consumed oil properties in the AVL Excite Piston and Rings Module.

Name: SAE 10W-40

Characteristics

	atmospheric pressure			200MPa
	@ 40°C	@ 100°C	HTHS[1)]	@ 160°C and $10^6 s^{-1}$
Dynamic Viscosity (mPa.s)	70.435	12.286	5.177	4.520
Density (kg/m^3)	864.000	864.000	864.000	864.000
Specific Heat Capacity (J/kgK)	2083.000	2083.000	2083.000	2083.000
Thermal Conductivity (W/mK)	0.140	0.140	0.140	0.140

1) High temperature and high shear rate, 150°C and $10^6 s^{-1}$

In Table 2.2 the oil properties which is used when running the AVL Excite oil consumption model is listed. SAE 10W40 grade oil is selected for oil consumption calculations.

The oil consumption model calculated 2000 rpm rotational speed for the crank shaft. Cylinder deactivation is conducted for part load working conditions. Therefore, the power and torque values should be same for normal working mode and skip cycle cylinder deactivation. However, as the oil consumption model is a single cylinder model, variable stroke volume cylinder deactivation cannot be a working condition for it. In order to get an indication for fully shut off cylinder oil consumption the thermodynamic data of skipped cycle in other words the last 4 stroke thermodynamic data of NS skip cycle working mode is used when calculating the lube oil transportation. NS means of 8 strokes working condition of skip cycle method. As previously discussed, skip cycle is anticipated as partial deactivation of the cylinder. NS demonstrates that the first cycle gives the useful work and second cycle shows fully deactivated cylinder characteristics.

Comparing the power and torque of normal working mode and cylinder deactivation condition, in both case 1.46 kW power and 7 Nm torque were achieved. Consequently, both working conditions are a good example of a back to back comparison of oil consumptions for both normal working mode and cylinder deactivation working strategy.

In the oil consumption model, constant 1 bar atmospheric crank case pressure and swirl ratio of 1.2 are assumed. Also constant material temperatures are assumed.

When calculating oil consumptions for different working conditions, liner surface temperatures, piston temperatures and ring temperatures are kept as same.

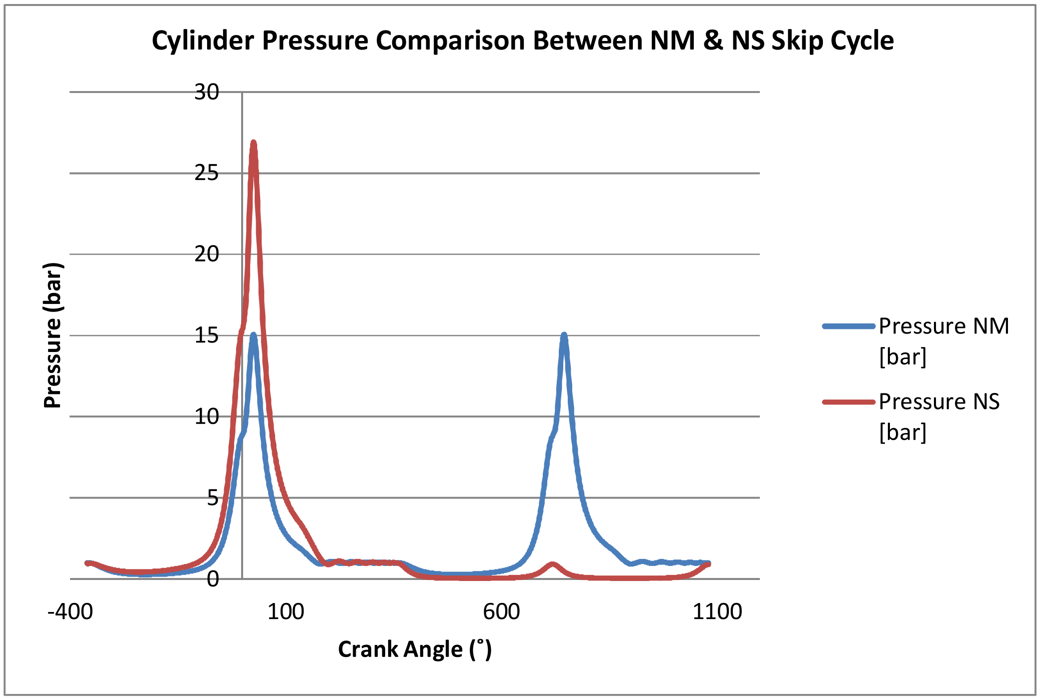


Figure 2.12 : Cylinder pressures of normal operating mode and 8 stroke skip cycle mode.

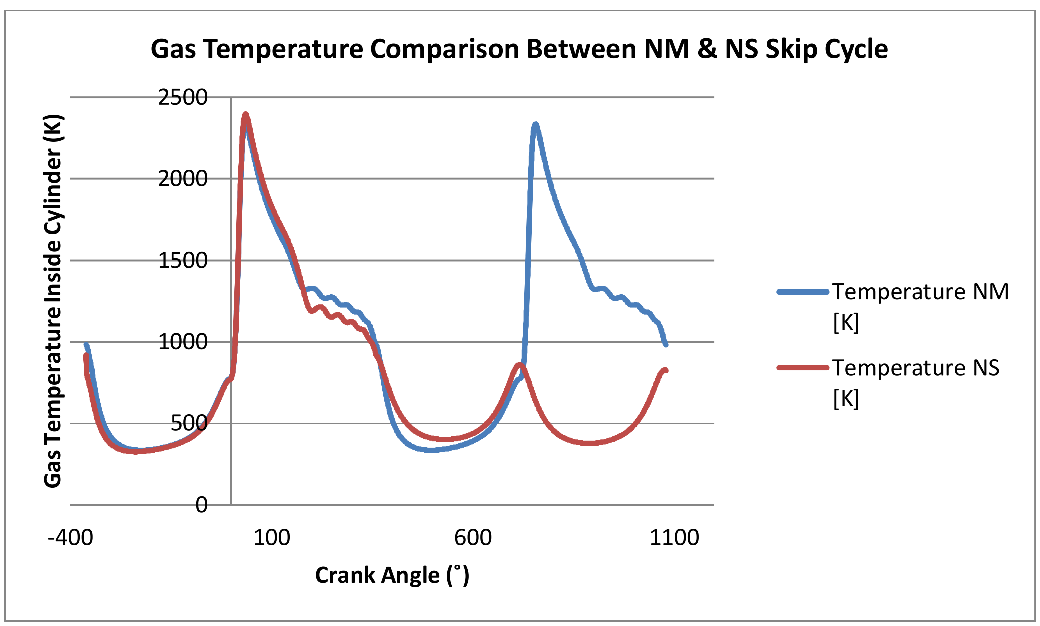


Figure 2.13 : Gas temperatures of normal operating mode and 8 stroke skip cycle mode.

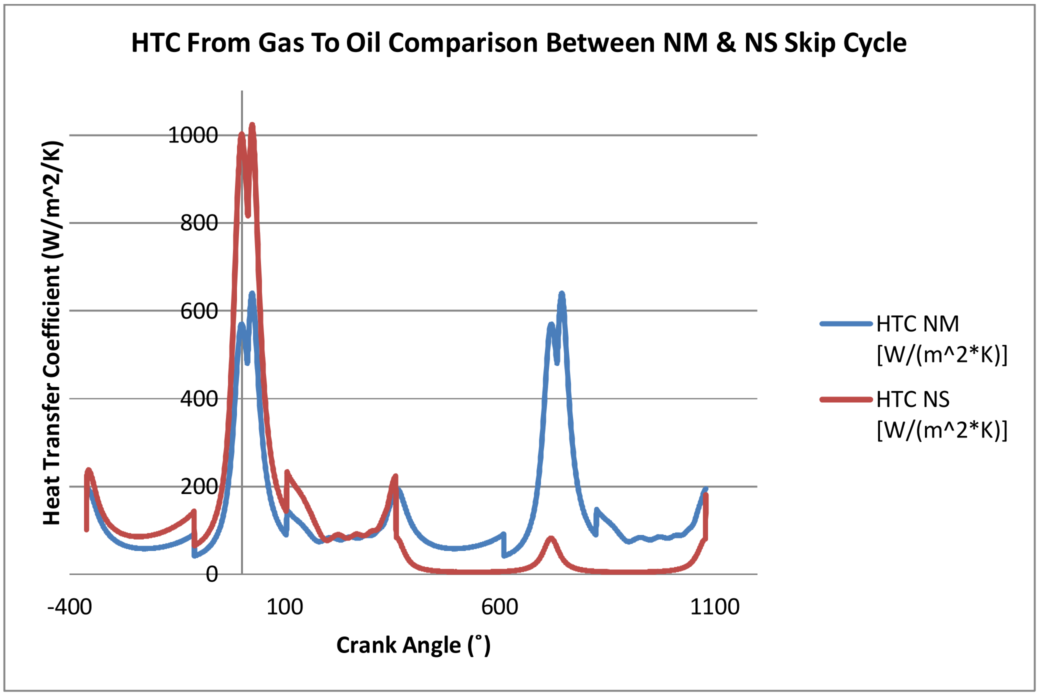


Figure 2.14 : Heat transfer coefficients from gas inside the cyliner to oil on liner surface of normal operating mode and 8 stroke skip cycle mode.

The thermodynamic data presented in Figure 2.12, Figure 2.13 and in Figure 2.14 is produced in Ricardo Wave. In normal operating mode all pressure, temperature and heat transfer coefficient values are lower as expected in the first 720˚ crank angle. However, looking at overall, the power and torque is same at the end of 8 stroke.

The above figures from 2.12 to 2.14 contains all information about the assumptions taken into consideration during oil transport calculations. The data corresponding to the crank angles between -360˚ and +360˚ is for the comparison of a conventional working mode of a cylinder and active cylinder of a cylinder deactivation applied engine. The rest of the data which corresponds to the crank angles between +360˚ and +1080˚ is for the comparison of a conventional working mode of a cylinder and passive or closed cylinder of a cylinder deactivation applied engine. In overall, from -360˚ to +1080˚, the figures can be assumed to be the comparison of a conventional working cylinder and partially deactivated cylinder. Partial deactivation can be paraphrased as skip cycle mode.

3. RESULTS AND DISCUSSION

3.1 Total Oil Consumption

The total oil consumptions were calculated for a period of 1440˚ crank angle which corresponds to 4 revolutions of crank shaft, 8 stroke or 2 thermodynamic cycles of the engine. The reason of defining the oil consumption period as 2 thermodynamic cycles is that for the skip cycle cylinder deactivation in one cycle, combustion occurs and in the following cycle valves are closed and combustion does not occur. Therefore, for a back to back comparison 2 consecutive thermodynamic cycles were taken into account for normal operation mode and variable stroke volume cylinder deactivation. Shortly, partial cylinder deactivation oil consumption, which is the skip cycle method, cannot be compared with full deactivated cylinder, which is the variable stroke volume (vStroke) cylinder deactivation, and conventionally working cylinder oil consumptions if only 4 strokes were analyzed.

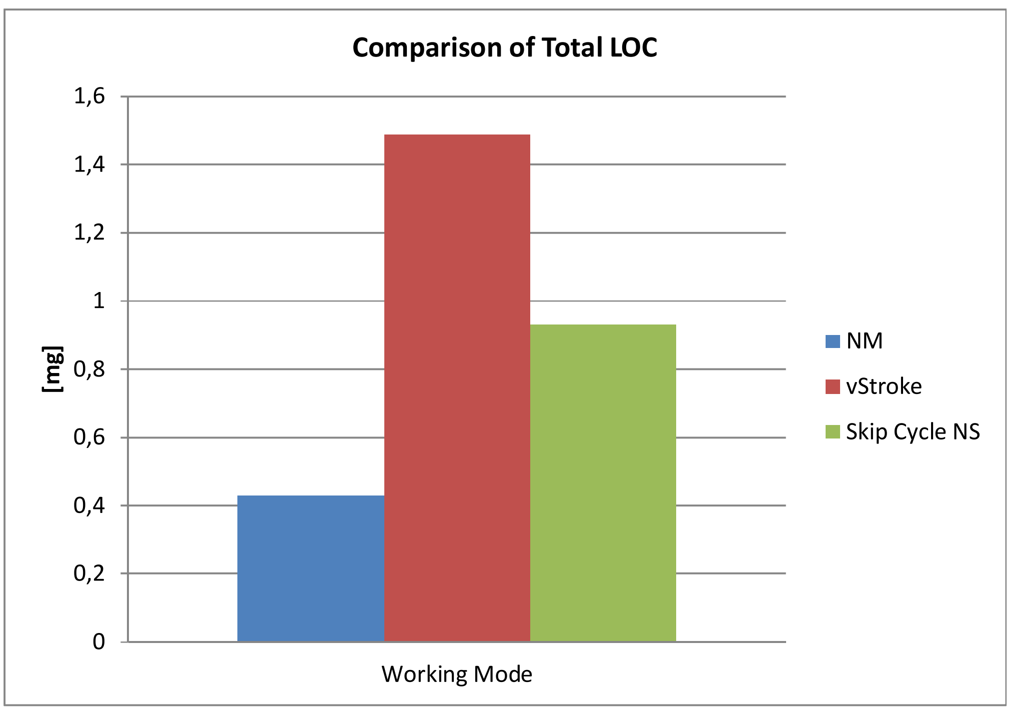
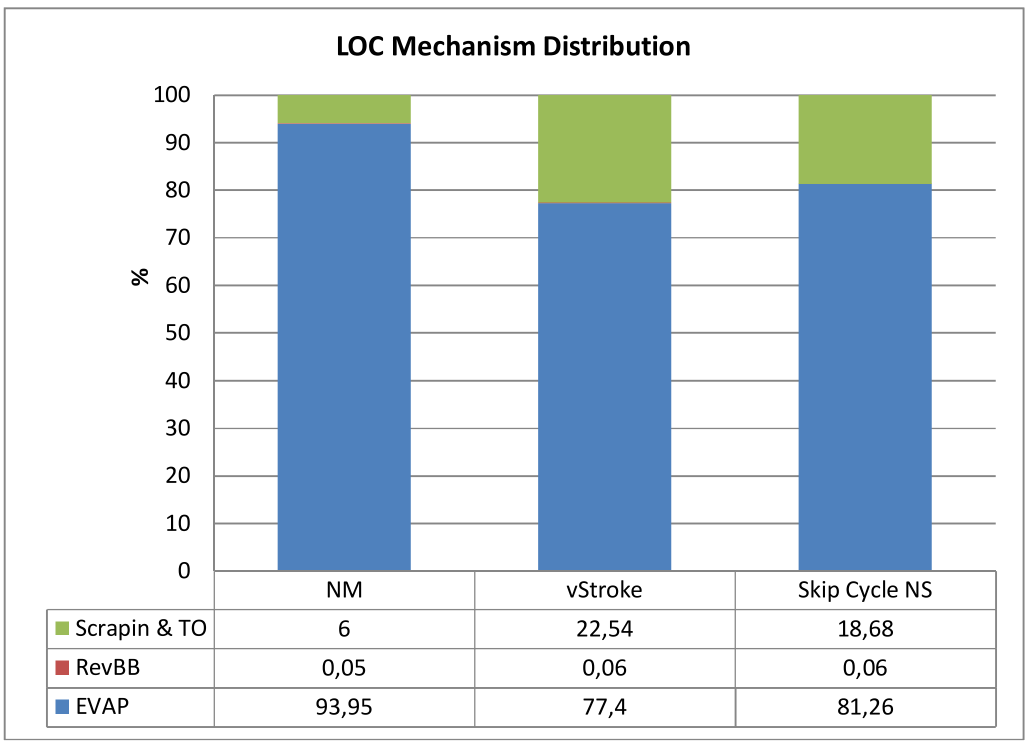


Figure 3.1 : Total oil amount transported into the combustion chamber at the end of 4 crank shaft revolution or in period of 1440˚ crank angle.

The biggest amount of oil consumption was detected during variable stroke volume cylinder deactivation which has vacuum pressure in the cylinder. Although the pressure curves shown on Figure 2.12 are passing through positive values, 'vacuum pressure' term was intentionally used as the curves are showing absolute pressure values and stay below atmospheric pressure when valves are closed. The main reason underlying full cylinder deactivation of variable stroke volume method has the biggest oil consumption is that the mass fluxes of the oil transported into the cylinder for all kind of transportation mechanisms are much higher when combustion does not occur in the cylinder. The details are discussed on the following pages under related oil consumption mechanisms.

Also it should be noted that the oil evaporation is the most significant contributor of the total oil transportation into the cylinder. The least important oil transportation mechanism is the reverse blow by effect, which can be neglected as it is seen in Figure 3.2.



	NM	vStroke	Skip Cycle NS
Scrapin & TO	6	22,54	18,68
RevBB	0,05	0,06	0,06
EVAP	93,95	77,4	81,26

Figure 3.2 : Contributions of oil transportation mechanisms in different working conditions.

As a reminder, variable stroke volume cylinder deactivation condition is assumed as the repetition of the skipped cycle of the partially deactivated cylinder. The dominating oil consumption mechanism is the oil evaporation. The distributions of the contributions of oil consumption mechanisms are expected to be similar for both normal operating mode and the firing cycle of the …NS… cylinder deactivation strategy (partially deactivated cylinder). However, as the total oil mass flux increases dramatically when firing does not occur, the contributors distributions are quite similar for vStroke working condition and skip cycle NS working condition.

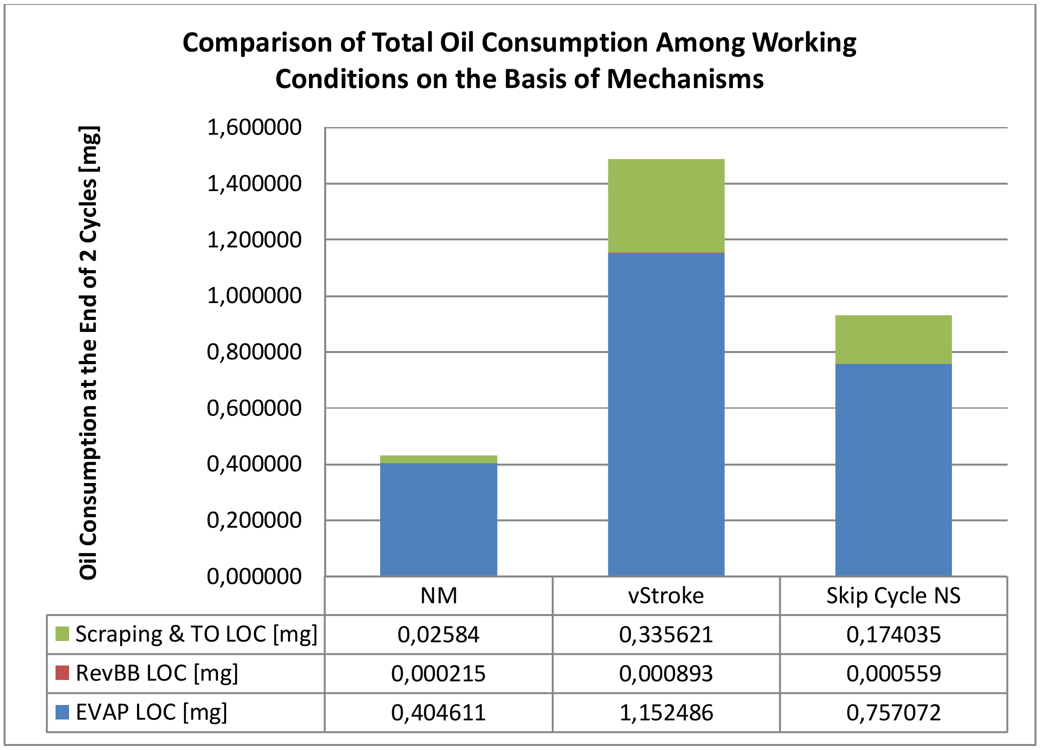


Figure 3.3 : Oil consumptions depending on the working conditions on the basis of transportation mechanisms through the interface of piston assembly and cylinder wall.

In the following sections the oil consumption mechanisms are discussed in details. Transferred into the cylinder oil amounts are shown on graphs on the basis of crank angle. The firing or first top dead center and scavenging or second top dead center are demonstrated as F-TDC and S-TDC respectively.

3.2 Oil Consumption Due To Evaporation

Recalling Equations 2.10 and 2.12, the main parameters which affect the evaporated oil amount are material transmission coefficient, diffusion coefficient and oil concentrations at the cylinder wall oil film surface and inside the cylinder. The evaporation rate is affected by the oil concentration gradient. Oil concentration gradient is directly proportional with the partial oil vapor pressure gradient from the liner surface to the center of the cylinder.

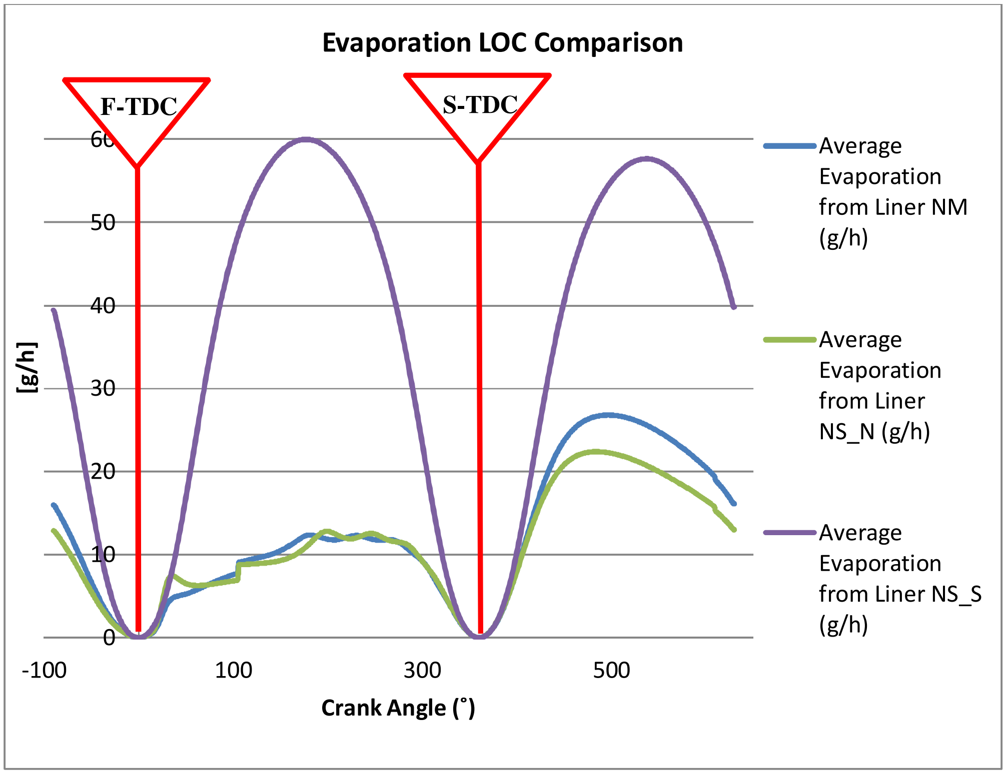


Figure 3.4 : Oil mass flux from liner surface to cylinder due to evaporation among the working conditions of NM, NS_N and NS_S.

In the not firing condition oil evaporation rate is extremely high. And as expected, for the firing cycles of normal operation mode and the firing part of NS skip cycle mode, the evaporation rates show similar trend. At this point the mechanism can be questioned for not firing cycle.

Comparing the not firing cycle; in other words the skipped cycle, with the firing cycles, mainly vacuum pressure is present in the not firing cylinder. This vacuum increases the partial vapor pressure gradient of the oil. Also material transmission coefficient and diffusion coefficient may increase with vacuum pressure. However, temperature does not have a direct effect on evaporation. It affects oil film surface vapor pressure; and, evaporation coefficients can be affected by temperature. Oil film temperatures were found as changing in a small interval and are similar for all three conditions. Therefore, it can be concluded as the main thermodynamic property changing the oil evaporation rate for those scenarios is the cylinder pressure.

In addition to direct effecting parameters of evaporation, oil film thicknesses also change when combustion is skipped. Although the oil film thicknesses are similar at the thrust side of the piston assembly, at the anti-thrust side the situation is different. When combustion does not occur and no load condition happens, the oil film thickness increases at the anti-thrust side of the piston assembly. Therefore, in the skipped cycle cylinder more oil amount is present at the liner surface which is a potential for the higher evaporation rates.

3.3 Oil Consumption Due To Reverse Blow By Gas Flow

Again in not firing condition; in other words the skipped cycle, the oil mass flux into the cylinder is the biggest among those three scenarios as expected.

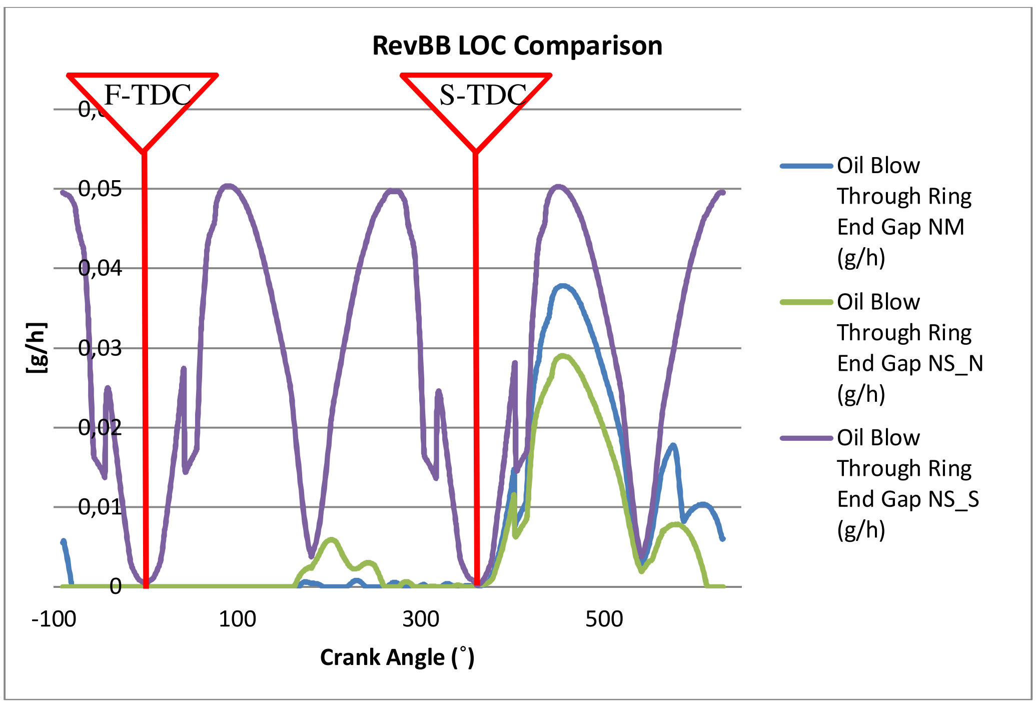


Figure 3.5 : Oil mass flux transported into cylinder due to reverse blow by gas flow among the working conditions of NM, NS_N and NS_S.

Referring Equation 2.15, oil transfer due to reverse blow by gas flow only occurs when the pressure between top ring and second ring gets bigger than the pressure inside the cylinder. Looking at the overall of the thermodynamic cycles, when cylinder is shut down and no combustion occurs, there is vacuum pressure present in the cylinder. As per the model assumptions, 1 bar constant crank case pressure is defined. This means, there is continuous reverse blow by gas flow into the cylinder.

Also looking at the other effecting parameters, as not firing cycle gas temperatures are much lower compared to the firing cycles, the actual area at the ring end gap is bigger and the width of the ring running face is smaller in the not firing cylinder. This is basically because the higher temperature is the higher expansion rate and vice versa. In addition to the geometric factors, as there is minor change in the oil film temperature, the change in oil properties such as viscosity and density would be minor.

3.4 Oil Consumption Due To Scraping And Throw Off

As same as the previous two oil consumption mechanisms discussed in section 3.2 and section 3.3, in not firing condition; in other words the skipped cycle, the oil mass flux into the cylinder is the biggest among those three scenarios as expected.

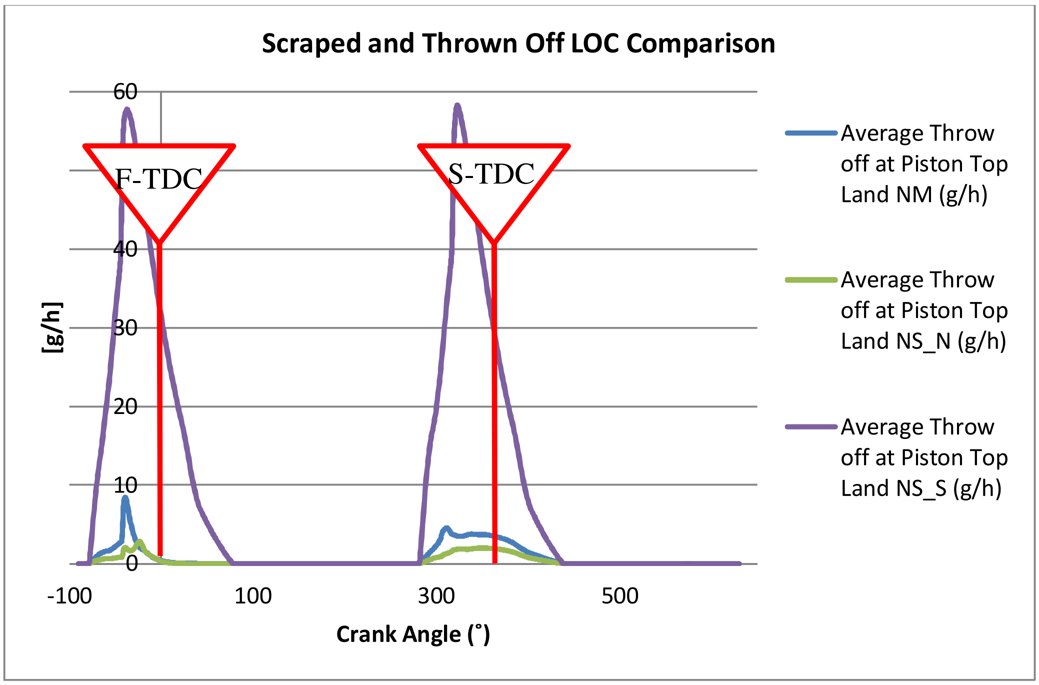


Figure 3.6 : Oil mass flux transported into cylinder due to piston scraping and top ring throw off.

Both scraping and throw off mechanisms basically stand on inertial forces. This is why the oil mass flux has the maximum value just before the piston reaches to top dead center in Figure 3.5. During the reciprocating motion of piston assembly, piston crown top edge and top ring scrapes some oil from the oil film at the liner surface. Scraped oil accumulates at the top surfaces of the first ring and the piston. While piston assembly is getting closer and closer to top dead center, its velocity decreases. At the top dead center it stops for an instant and starts travelling back to the bottom dead center. The deceleration of the piston assembly creates the inertial force that throws the accumulated oil on piston and top ring surfaces into the combustion chamber.

The accumulated oil amount on top of the first ring is the biggest when the combustion does not occur. It is mainly because of the thicker oil film present in not firing condition. Especially at the anti-thrust side of the piston, oil film thickness is significantly bigger when combustion does not occur. Load has significant influence on oil film thickness. The bigger load is the thinner oil film is. This phenomenon is basically related to hydrodynamic lubrication theory.

In general, top ring is found to be stable. However, for cycle skipped condition top ring moves rapidly at one instant for anti-thrust side only and looking at the full deactivation of the cylinder which is demonstrated under the working condition of variable stroke volume method the thrust side of the top ring can be said to be stable. Nevertheless, this minor movement of anti-thrust side of top ring in deactivated condition can be neglected when commenting on oil consumption. Top ring axial positions are shown in Appendix D for all working conditions.

a) b)

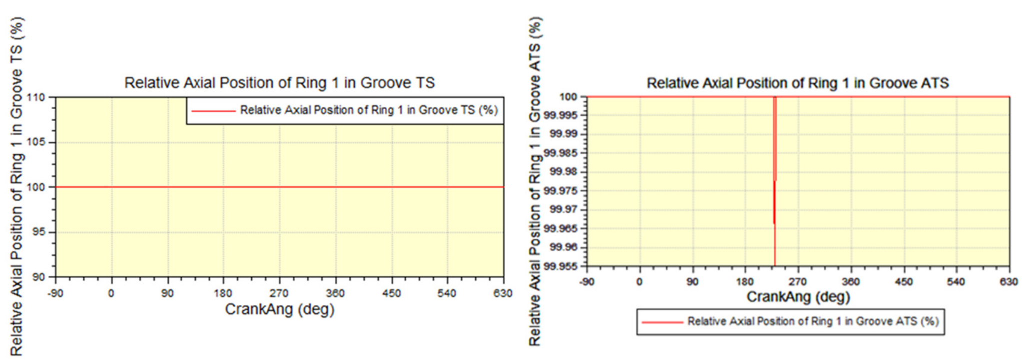


Figure 3.7 : a) Relative axial position of piston top ring at thrust side for vStroke condition. b) Relative axial position of piston top ring at anti-thrust side for vStroke condition.

As a summary, it is observed that the main contributor of the oil throw off and scraping mechanism is the change in the oil amount accumulating on top ring due to the increased oil film thickness on cylinder wall.

4. CONCLUSION AND RECOMMENDATION

Oil consumption has always been an important research area among automotive industry due to getting stringent emission regulations and customer demands. The efficiency of internal combustion engines has to be increased due to environmental concerns and to satisfy customer demands such as fuel economy and more powerful engines.

This thesis work investigated the oil consumption characteristics in a gasoline engine in different working cycles such as continuous combustion, continuous shut down cylinder, firing and not firing cycles consecutively. The investigation was done on a computer model and the model was based on a naturally aspirated spark ignition engine. Therefore, it can be said that this thesis combines the curiosity for oil consumption and for latest technology trends increasing engine efficiency.

It was found that although cylinder deactivation strategies increase engine efficiency and fuel economy they have adverse effect on oil transportation into combustion chamber through piston and cylinder wall interface. Taking the conventional operating mode as a baseline, partially deactivated cylinder mode has oil consumption of more than 2 times of the baseline and variable stroke cylinder deactivation mode; i.e. fully shut down cylinder, has oil consumption of more than 3 times of baseline.

This indication may lead to development of electronically controlled piston cooling jets. Electronically controlled piston cooling jets can adjust the oil flow which is sprayed at the bottom of the piston. When the cylinder is deactivated oil flow through the piston cooling jet can be reduced and so is the oil consumption. However, possible electronically controlled PCJ should respond fast enough to satisfy also skip cycle strategy.

The other focusing area can be the cylinder deactivation strategy itself on the basis of thermodynamic cycles to reduce the oil consumption. For example, the vacuum effect was analyzed for oil consumption in this work. However, positive cylinder pressure can reduce the oil consumption as well. Also, this study focuses on natural aspirated engine thermodynamics. Recently, engine downsizing is a hot topic and turbocharged engines are becoming common in the market. Furthermore, cylinder deactivation is applied on small volume turbocharged gasoline engines. Therefore, oil consumption of cylinder deactivated turbocharged engines can be a future work.

Experimental validation of the results that were discussed in this thesis would be another future study.

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APPENDICES

APPENDIX A

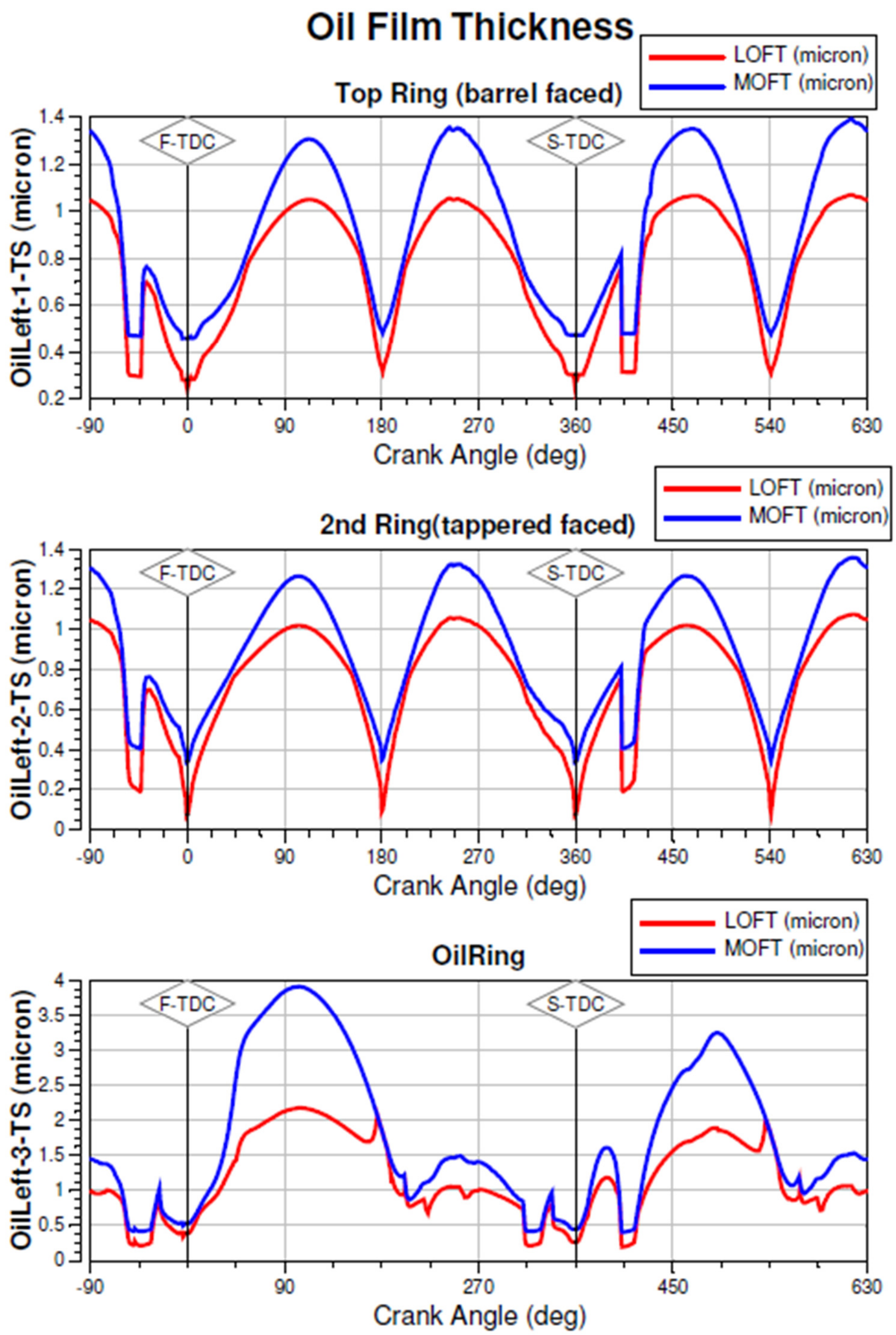

Figure A.1 : Oil film thicknesses over piston rings at the thrust side for normal mode.

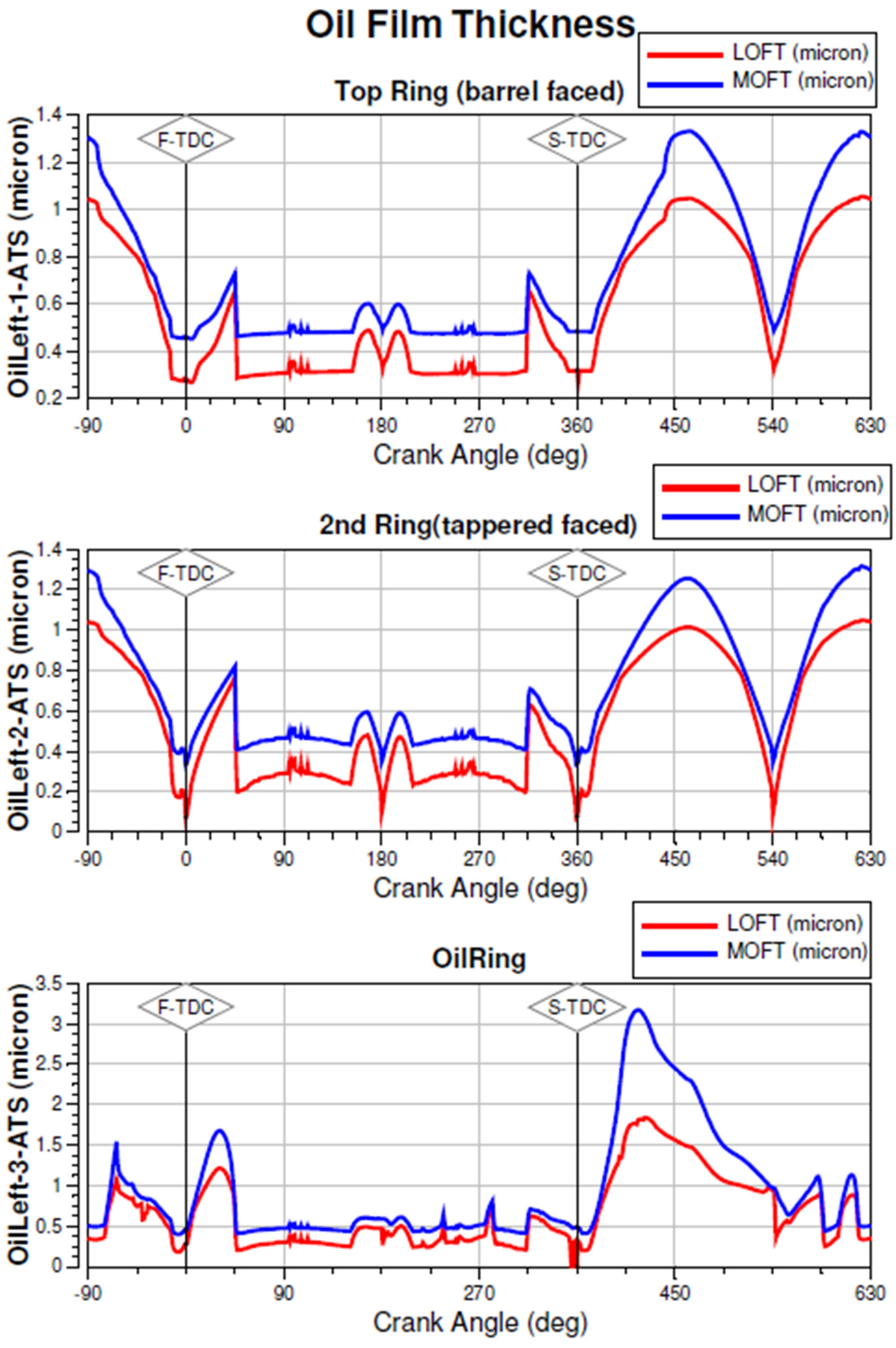

Figure A.2 : Oil film thicknesses over piston rings at the anti thrust side for normal mode.

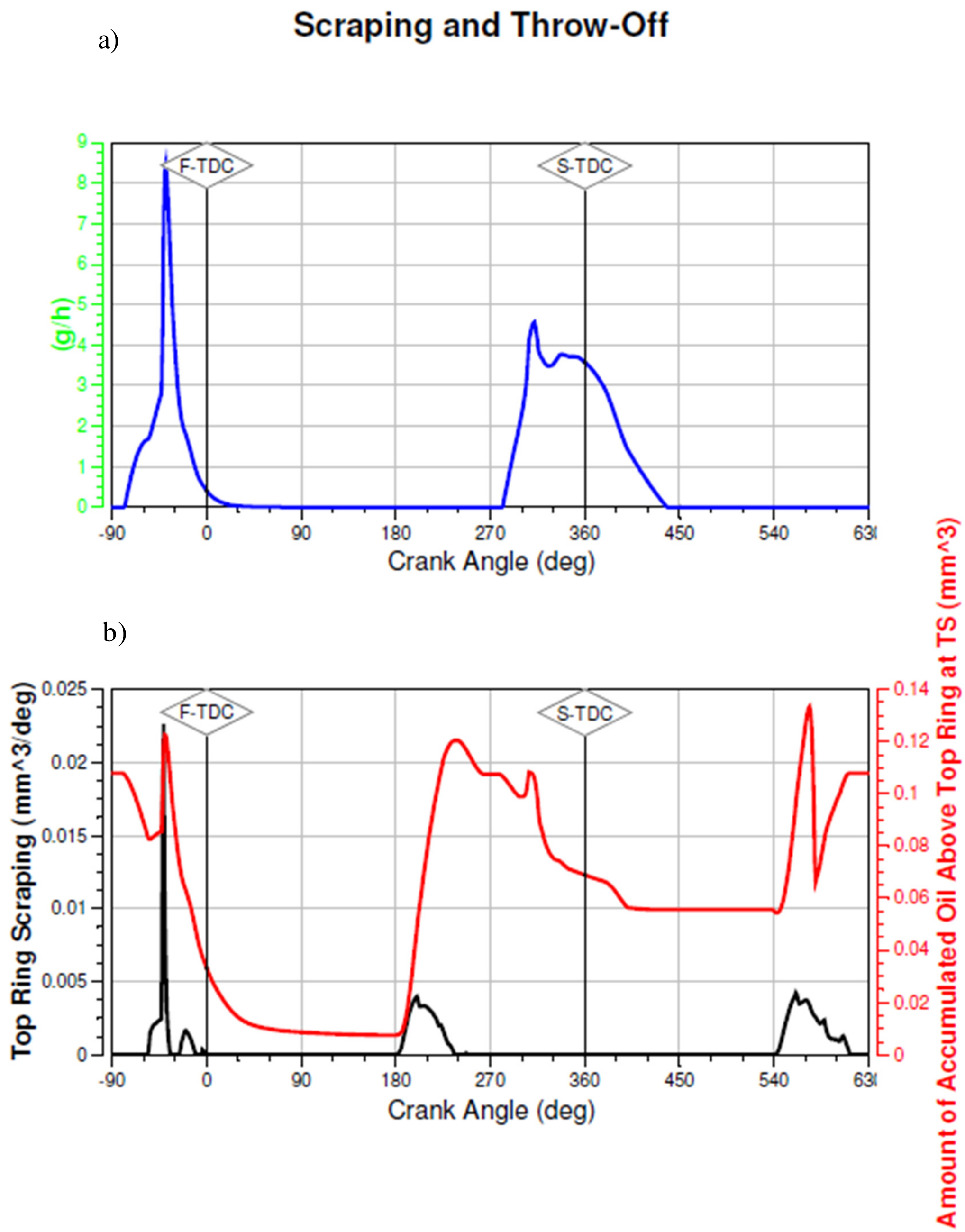

Figure A.3 : a) Scraped and thrown off oil mass flux during normal mode. b) Oil amount accumulated on top surfaces of piston and first ring.

a)

b)

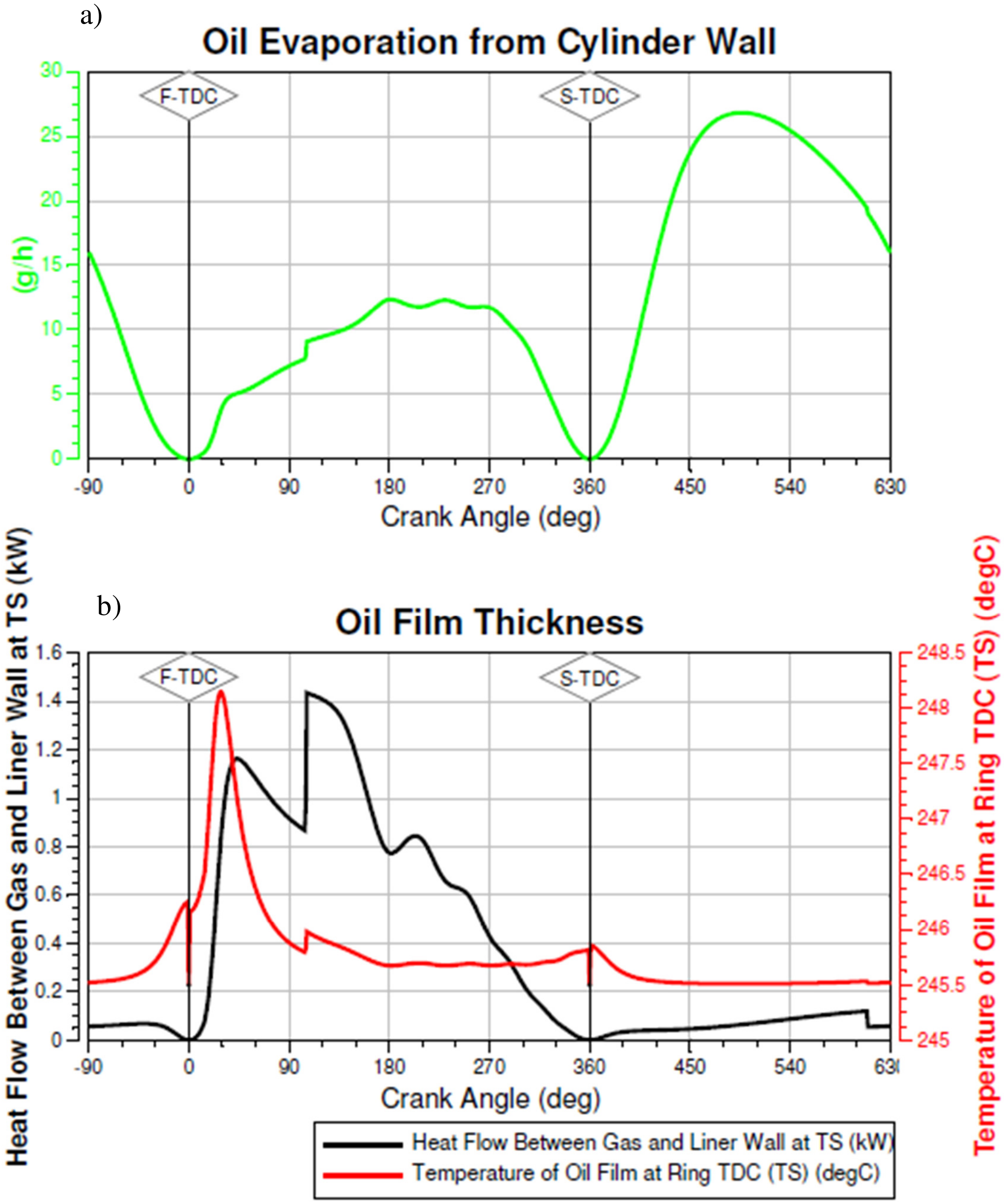

Figure A.4 : a) Oil mass flux during normal mode due to evaporation. b) Oil film temperatures.

APPENDIX B

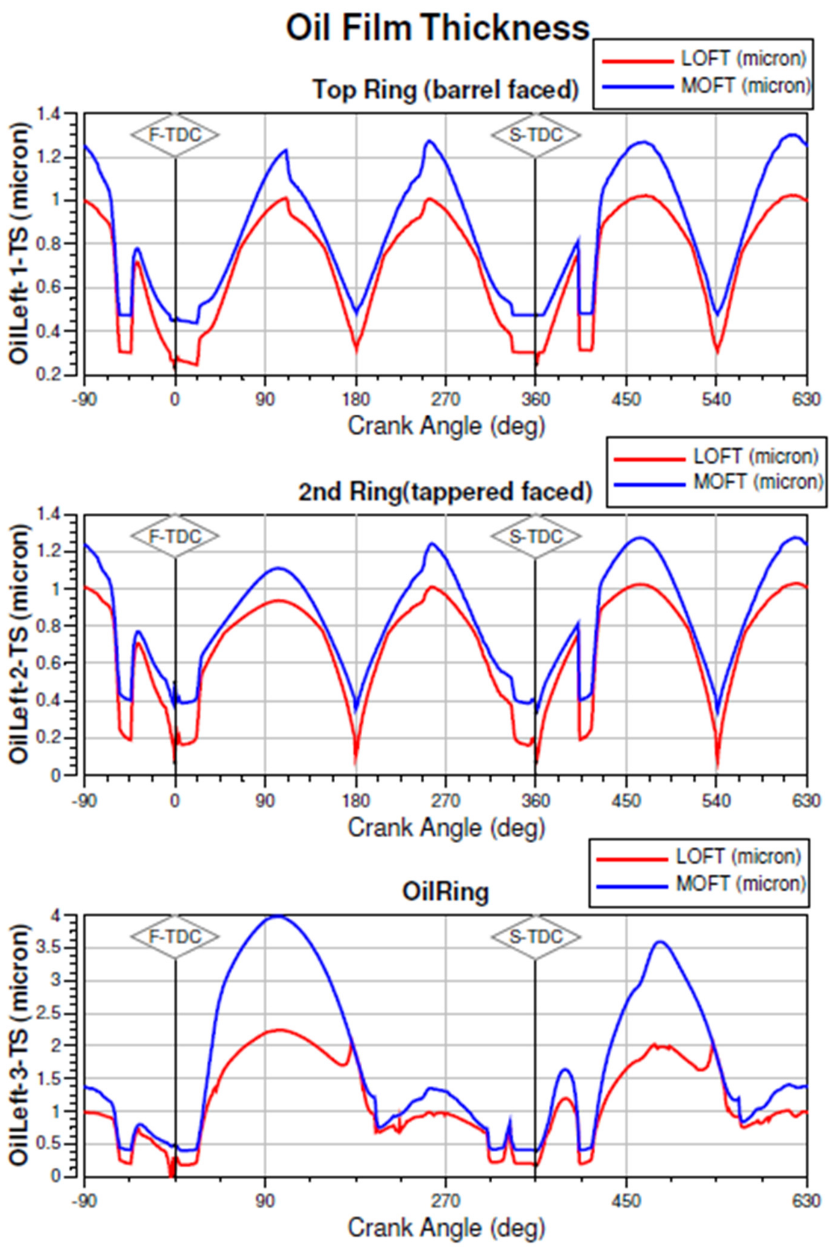

Figure B.1 : Oil film thicknesses over piston rings at the thrust side for the combustion portion of NS skip cycle mode.

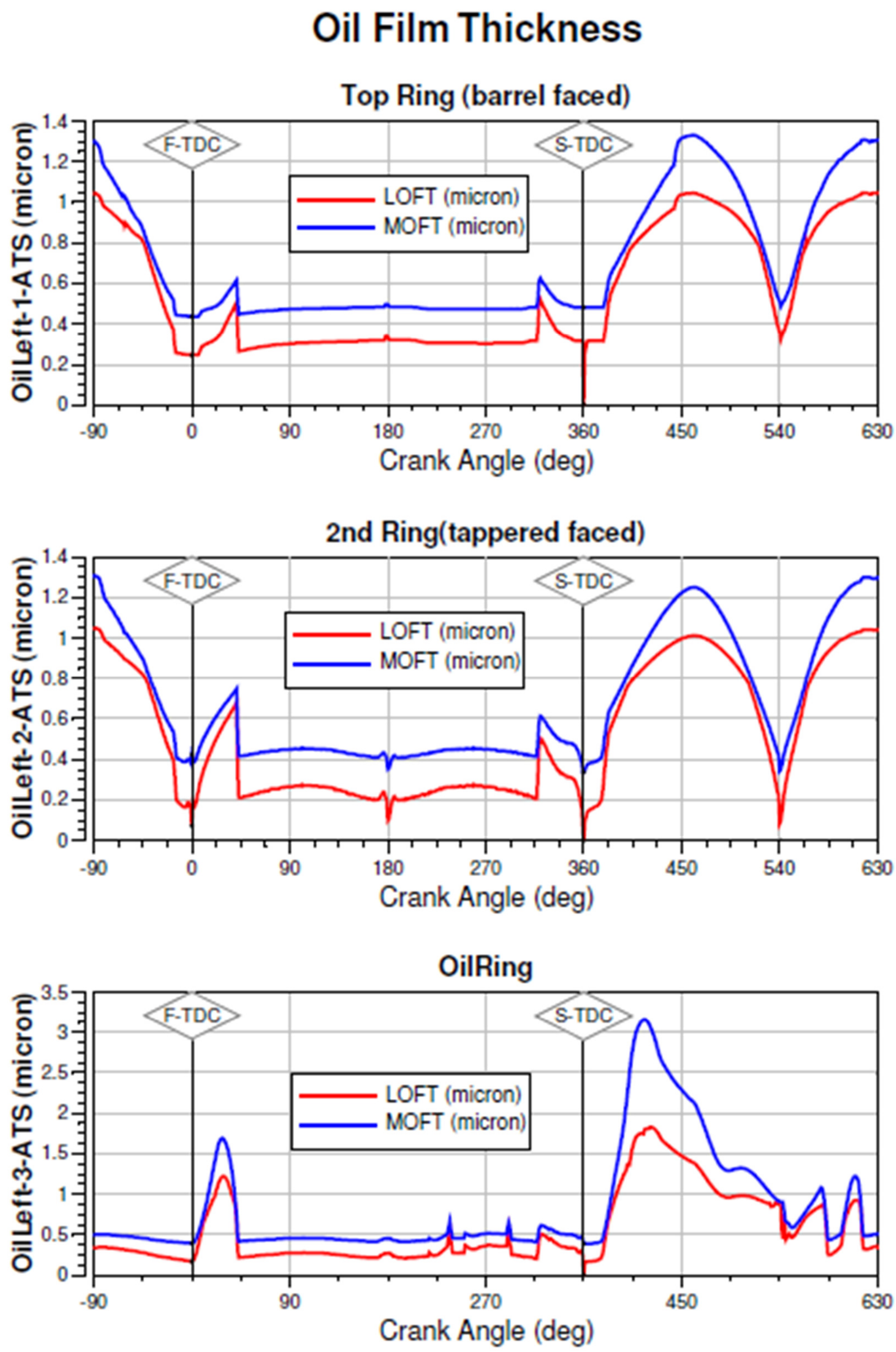

Figure B.2 : Oil film thicknesses over piston rings at the anti thrust side for the combustion portion of NS skip cycle mode.

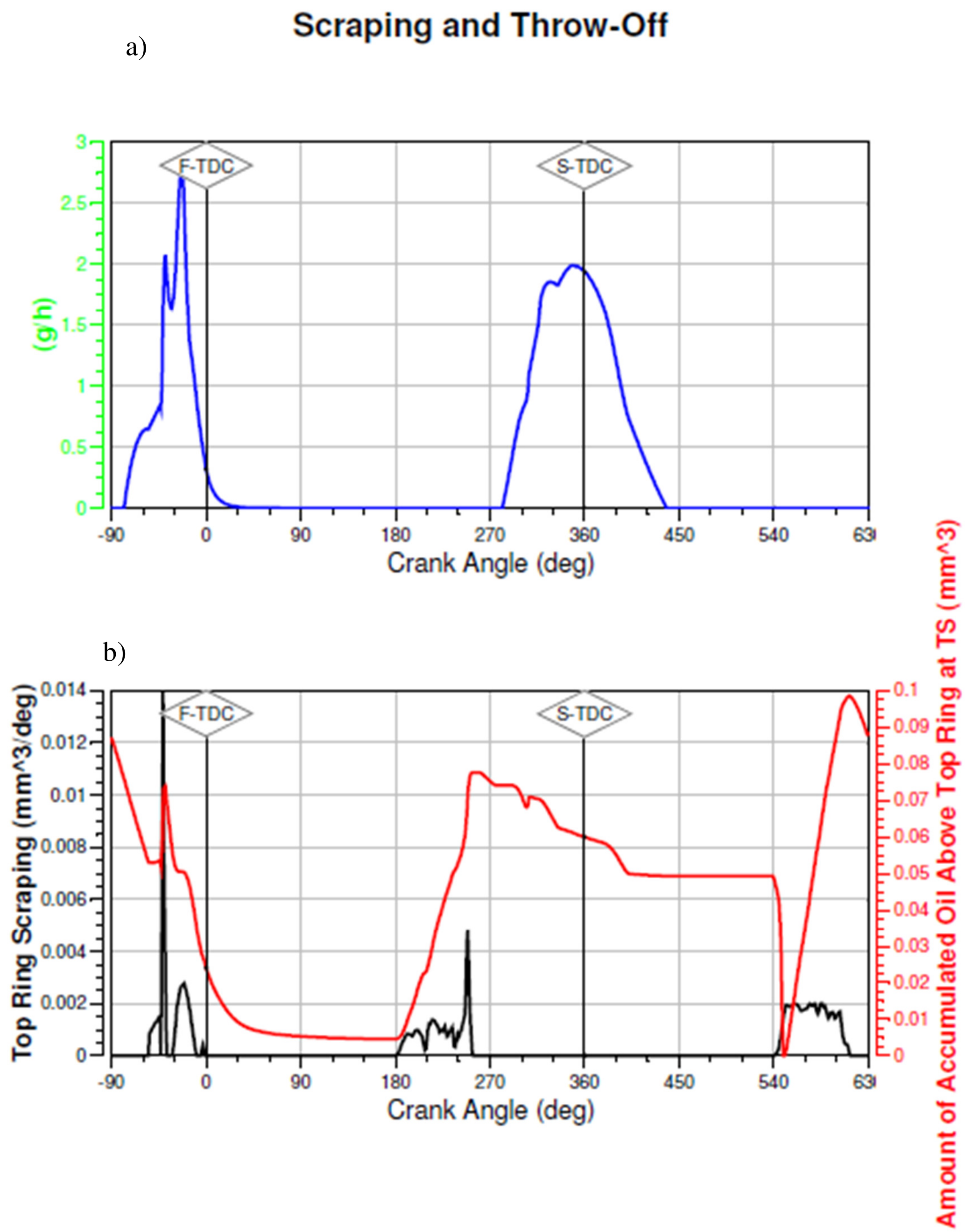

Figure B.3 : a) Scraped and thrown off oil mass flux during combustion portion of NS skip cycle mode. b) Oil amount accumulated on top surfaces of piston and first ring.

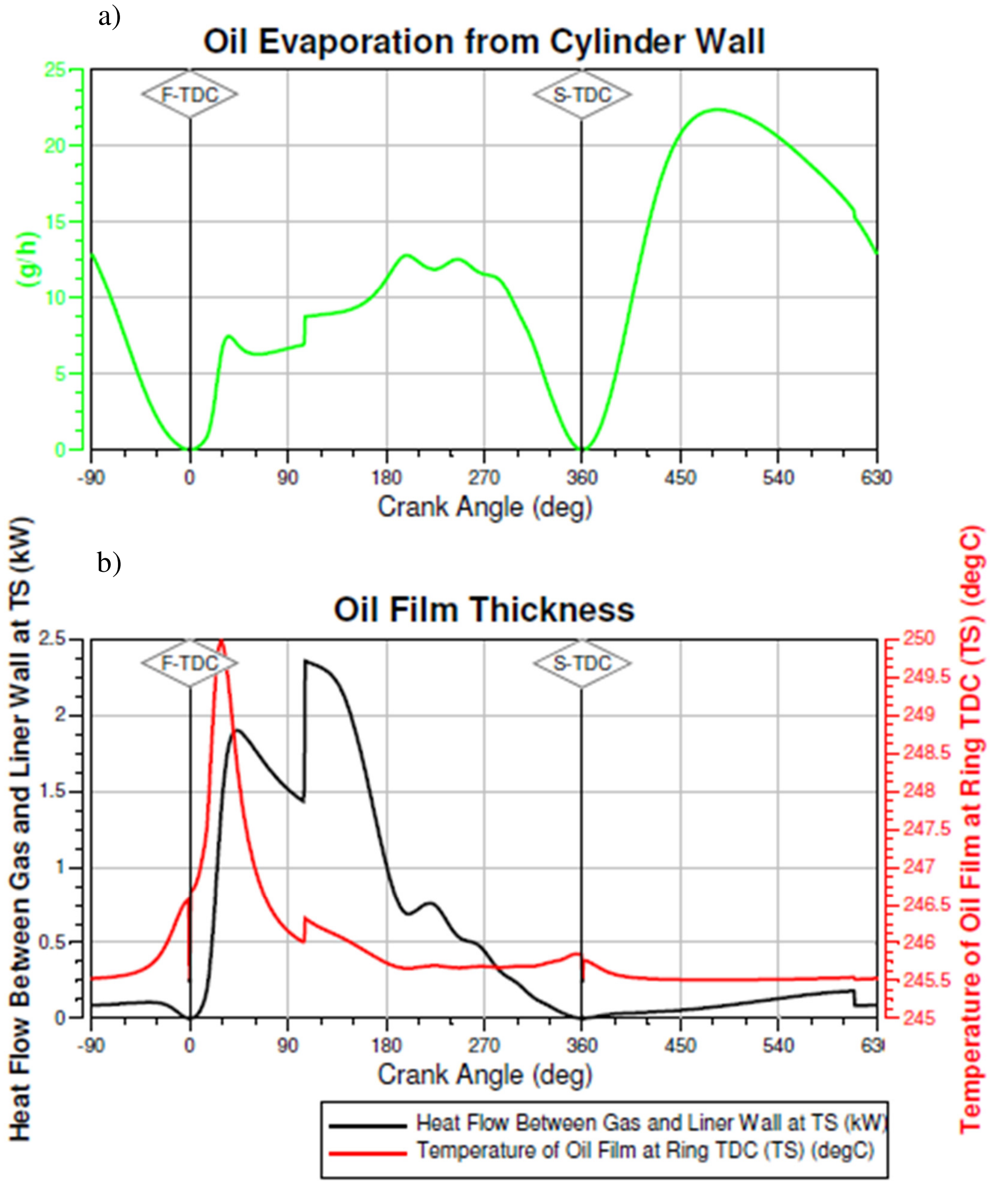

Figure B.4 : a) Oil mass flux during combustion portion of NS skip cycle mode due to evaporation. b) Oil film temperatures.

APPENDIX C

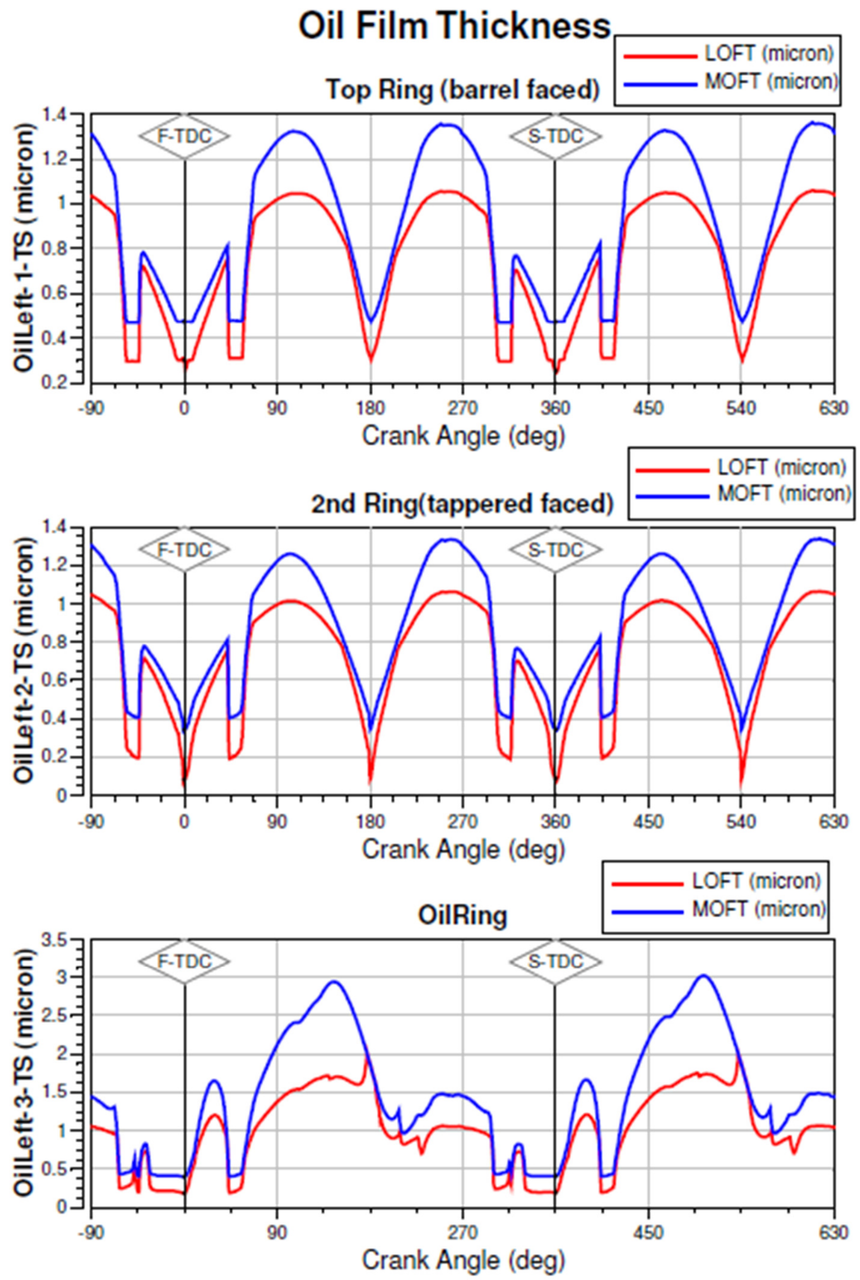

Figure C.1 : Oil film thicknesses over piston rings at the thrust side for the not firing portion of NS skip cycle mode.

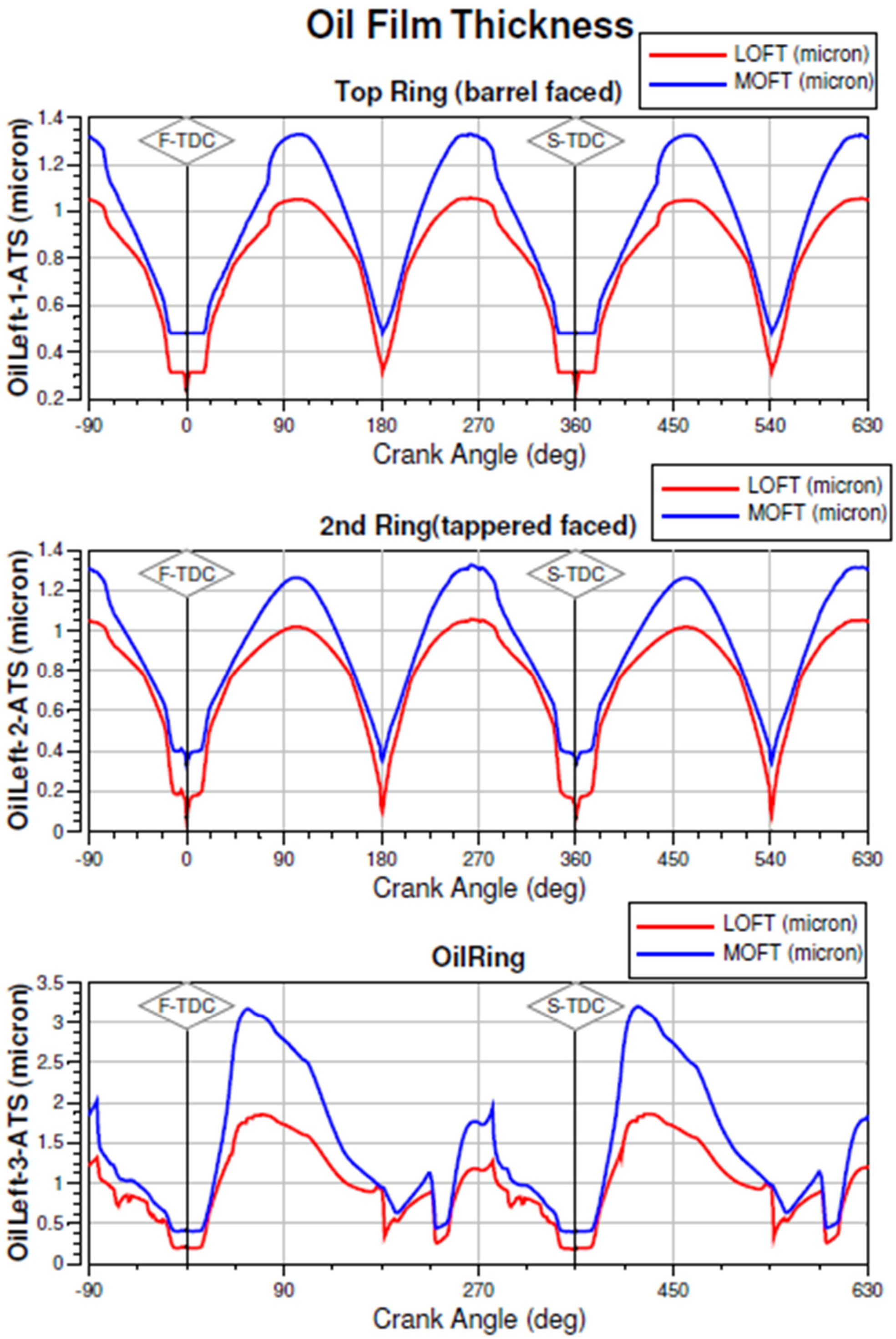

Figure C.2 : Oil film thicknesses over piston rings at the anti thrust side for the not firing portion of NS skip cycle mode.

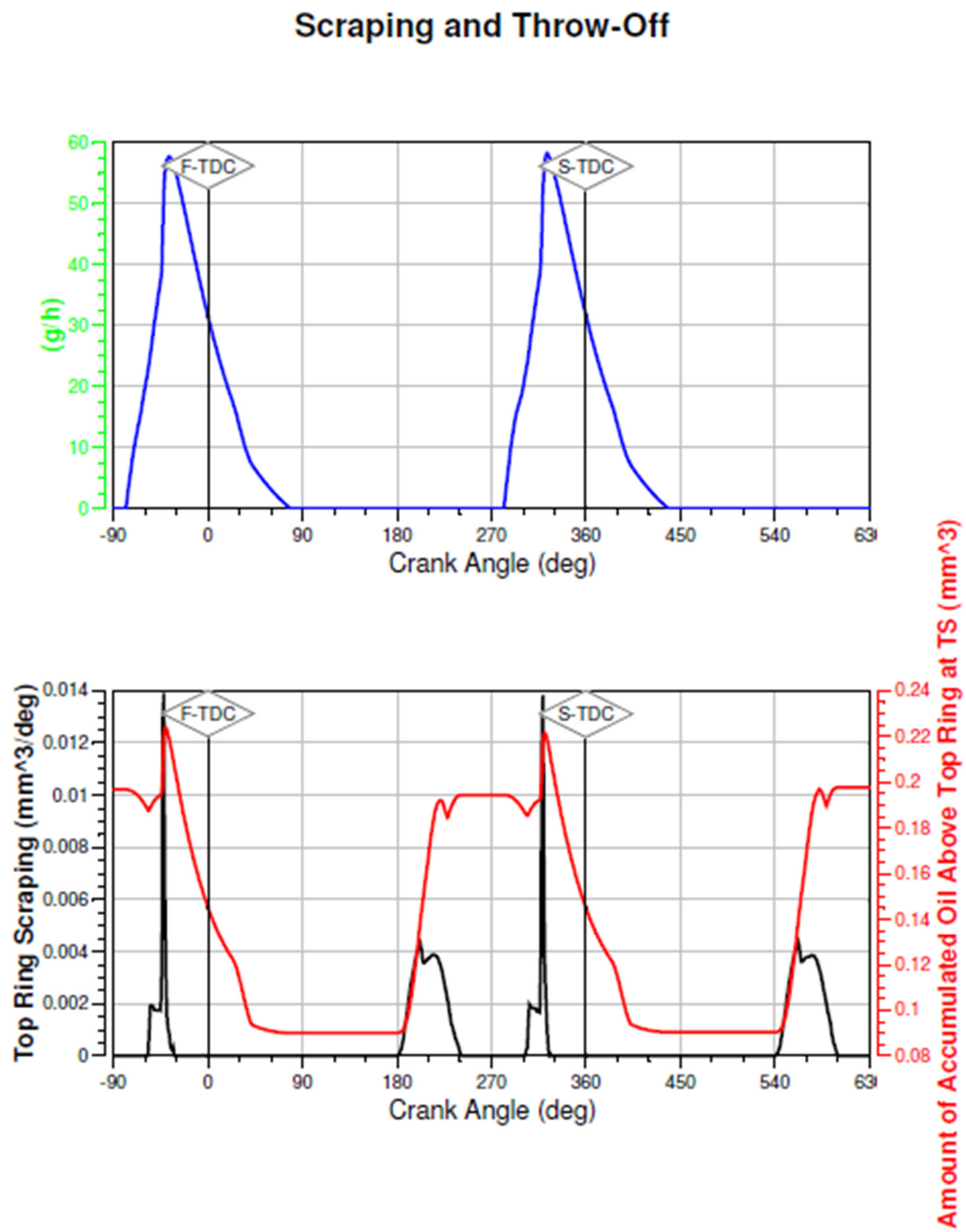

Figure C.3 : a) Scraped and thrown off oil mass flux during not firing portion of NS skip cycle mode. b) Oil amount accumulated on top surfaces of piston and first ring.

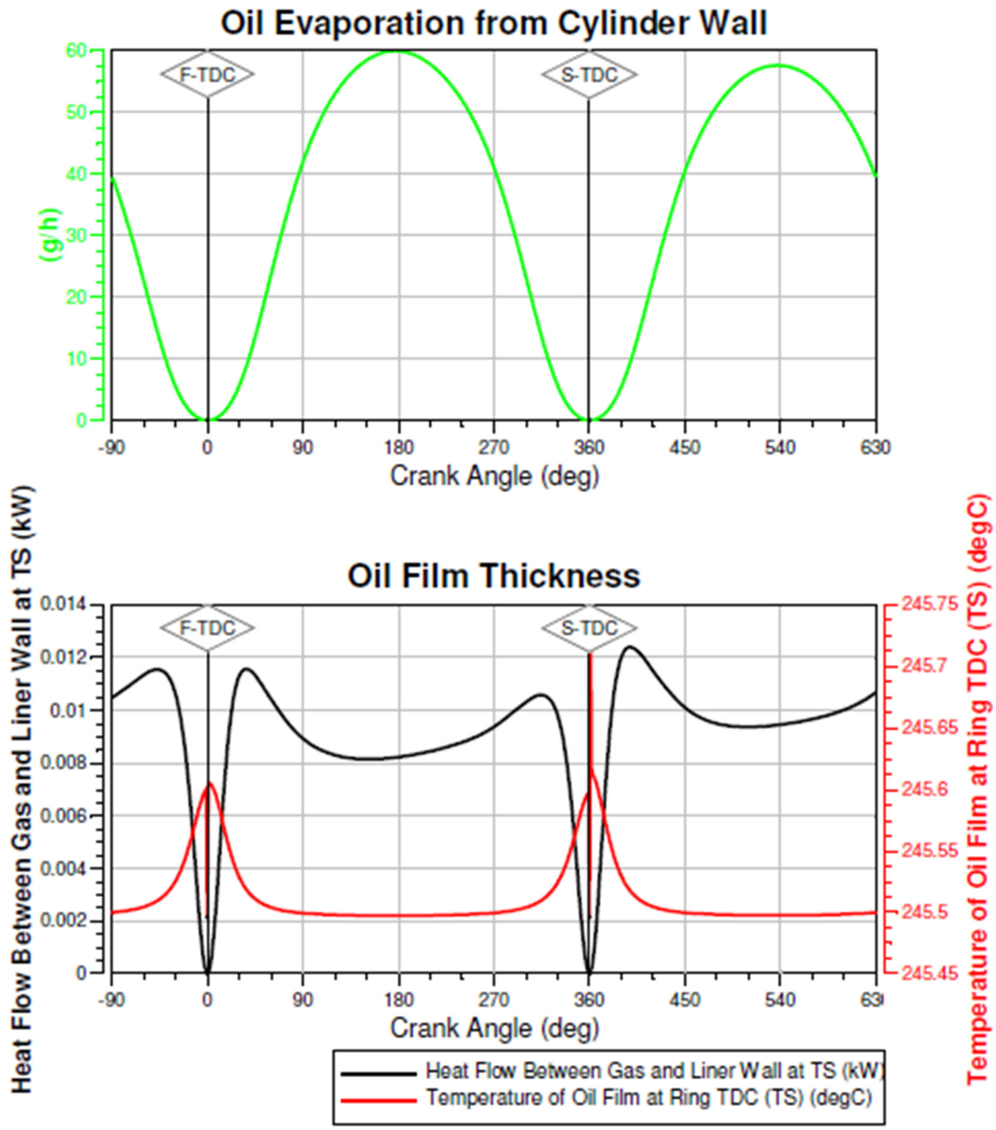

Figure C.4 : a) Oil mass flux during not firing portion of NS skip cycle mode due to evaporation. b) Oil film temperatures.

APPENDIX D

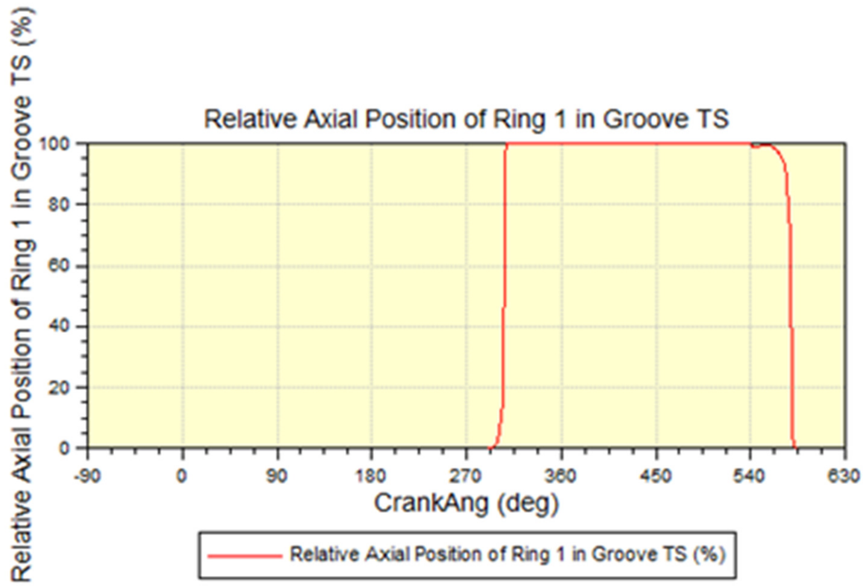

Figure D.1: Relative axial position of piston top ring at thrust side for NM condition.

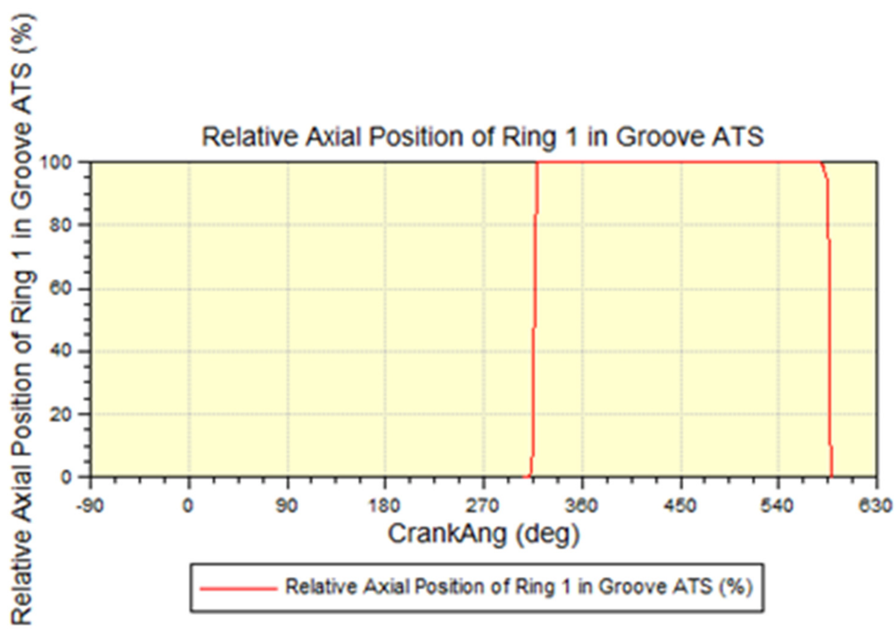

Figure D.2: Relative axial position of piston top ring at anti-thrust side for NM condition.

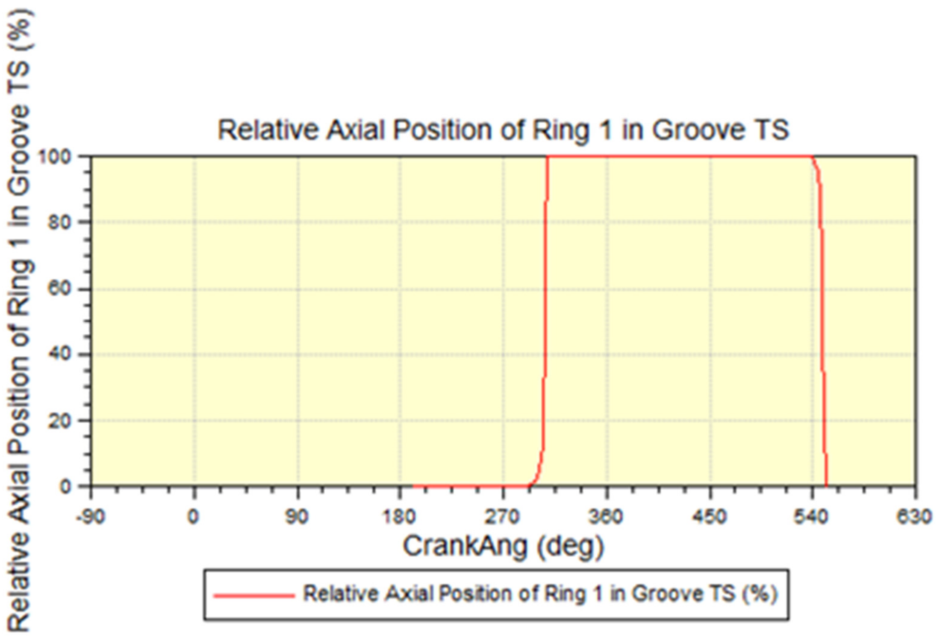

Figure D.3: Relative axial position of piston top ring at thrust side for NS_N condition.

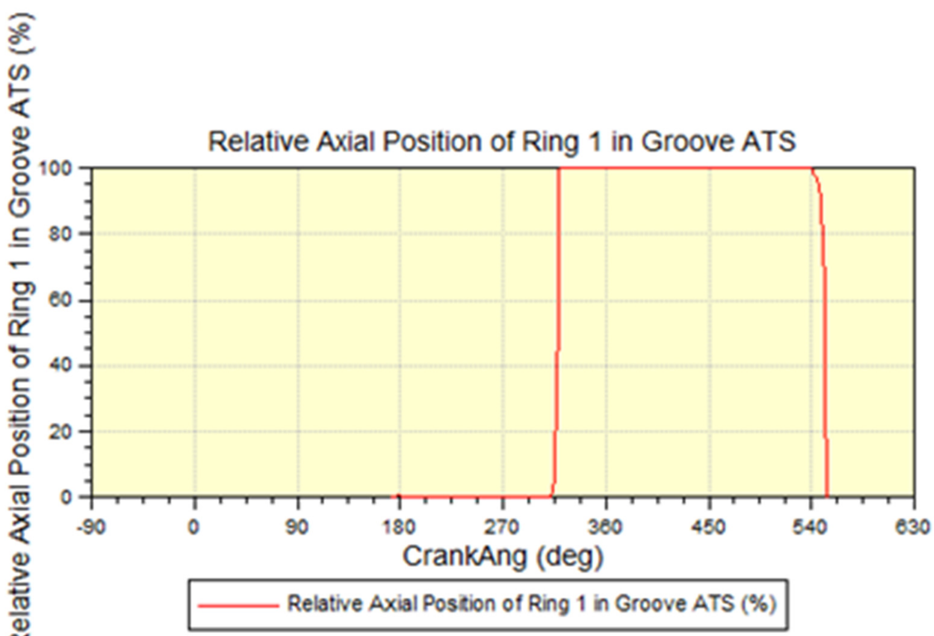

Figure D.4: Relative axial position of piston top ring at anti-thrust side for NS_N condition.

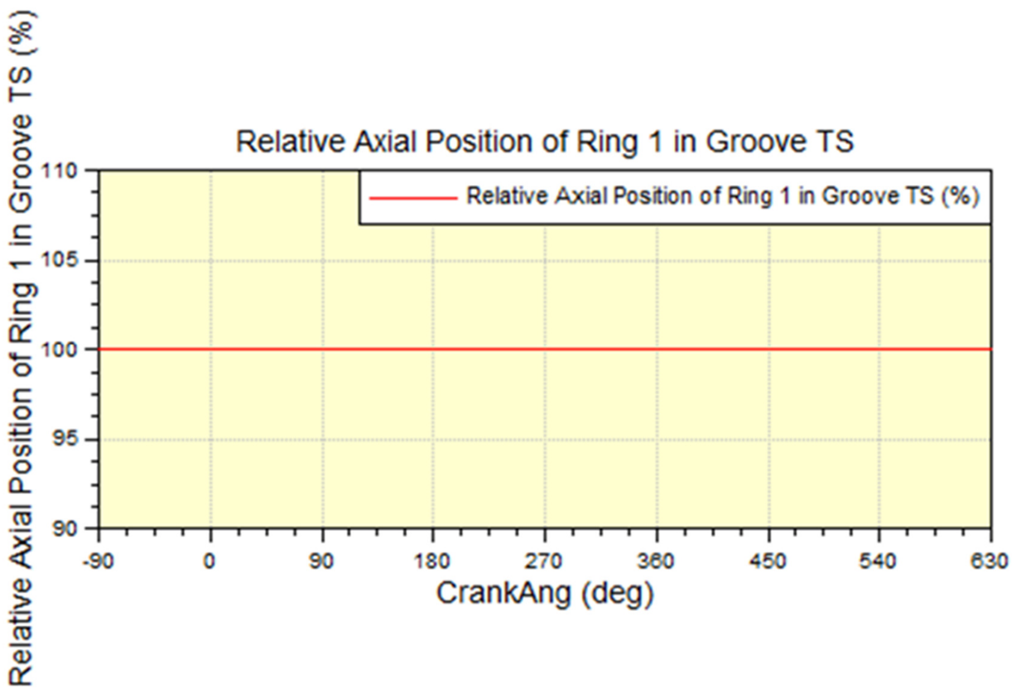

Figure D.5: Relative axial position of piston top ring at thrust side for NS_S condition.

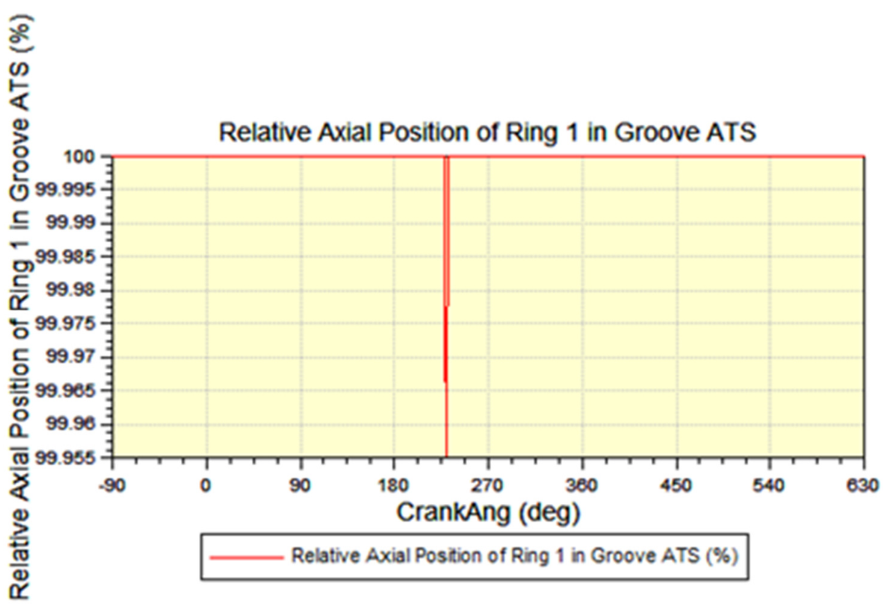

Figure D.6: Relative axial position of piston top ring at anti-thrust side for NS_S condition.

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- Ford Certificate of Excellence, L322 – 4.4L V8 Diesel Engine (3 March 2011)
- Ranked 1st in the 10th National Chemistry Olympics - Aegean Championship Award (2002)
- 10th National Chemistry Olympics 2nd stage winner and invitation for the National Team elimination (2002)